TRANSIENT STUDIES IN VERTICAL TUBE EVAPORATORS

by

DAVID BRANT MCGUIGAN, Lieutenant, United States Navy

B.S., United States Naval Academy
(1957)

SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS

FOR THE DEGREE OF NAVAL ENGINEER

AND THE DEGREE OF

MASTER OF SCIENCE IN NAVAL ARCHITECTURE

AND MARINE ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

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Department of Naval Architecture
and Marine Engineering, May 17, 1963

Certified by: ______________________________
Thesis Supervisor

Accepted by: ______________________________
Chairman, Departmental Committee
on Graduate Students

ABSTRACT

TRANSIENT STUDIES IN VERTICAL TUBE EVAPORATORS

by

Lieutenant David Brant McGuigan, USN

Submitted to the Department of Naval Architecture and Marine Engineering on May 17, 1963 in partial fulfillment of the requirements for the Master of Science degree in Naval Architecture and Marine Engineering and the Professional degree, Naval Engineer.

A considerable amount of work has been accomplished in the past in analyzing the steady state two-phase flow phenomena in vertical tube evaporators. Little has been done in the treatment of non-steady state conditions in forced-circulation systems. The transient responses in such systems have gained importance with increasing application of the monotube forced-circulation concept to the field of steam generation. A knowledge of the dynamic behavior of the forced-flow evaporator systems is of considerable importance in the control of once-through forced-circulation boilers. Such behavior dictates the feed control system and affects the design properties of the boiler as a whole.

To obtain a feeling for the transients involved in such systems a simplified test model was built. Steam of varying quality was generated at 20 psia in an electrically heated vertical evaporator constructed from 0.375" O.D. stainless steel tubing. Secured to the top of the evaporator, at the end of the uppermost heating unit, was a two foot length of pyrex tubing in which the boiling phenomena could be observed. Heat flow rate was varied from 0.5 KW to 2.0 KW. Weight flow rate was varied from 10 #/hr to 25 #/hr.

A total of fifty-three runs were conducted. The response of the system to four flow conditions was investigated: weight flow rate constant, step changes in heat flow rate (positive and negative) and heat flow rate constant, step changes in weight flow rate (positive and negative).

Total interface movement between two steady state conditions was determined through a correlation with peak outside wall temperature. Time and space variation in the commencement of nucleation and the end point of evaporation was related to the variation of outside wall temperature at the end of the generating section. Visual observations of the regimes of boiling under transient conditions were correlated with this time variation in outlet generator temperature.

Saturated and superheated interface movement under transient conditions in response to step change in both weight flow rate and heat flow rate were described by a general equational form:

$$\Delta L = \Delta L_{\infty}(1 - e^{-at})$$

The time constant, "a", of this equation is an indication of the time characteristics of the evaporator. It determines the initial slope of the transient response and is a strong function of weight flow rate and heat flow rate. Five regimes of boiling could be visually recognized in vertical tube evaporations: bubble, slug, slug-annular, turbulent-annular, and smooth annular.

In addition, the thesis introduces a method by which the empirical equations derived in the investigation can be employed in the development of transfer functions for use on the analog computer.

Thesis Supervisor: Ernst G. Frankel
Title: Assistant Professor of Marine Engineering

ACKNOWLEDGEMENT

The author is sincerely grateful for the association he has had with Professor E. G. Frankel, Thesis Supervisor. He is deeply indebted for his continued interest and guidance in the investigation.

This work could not have been successfully undertaken without the understanding and co-operation of Captain W. W. Braley, USN, Director of the U.S. Naval Boiler and Turbine Laboratory, Philadelphia, and his able staff who were of tremendous aid during the first phase of this investigation. The author is especially appreciative of the assistance given to him in Philadelphia by Mr. William I. Valentine in the development of the temperature measurement instrumentation and Mr. P. H. Wermuth in its construction. Mr. R. A. Szczepanski rendered incalculable assistance in the design of the test unit at NBTL.

Acknowledgement is extended to the staff of the EPL Laboratory, M.I.T., and especially to Mr. J. A. "Tiny" Calogerro, for without his assistance the construction of the apparatus for the second phase of the project would have been impossible. The author is most appreciative to Professor Peter Griffith for his conversations on the topic and to Mr. Charles G. McGuigan, Code 558-B, David Taylor Model Basin for his assistance in acquiring certain bibliographical material.

Lastly, the author is indebted to his wife, Kosh, who during his years of post-graduate study was a supplier of

encouragement and of understanding. She cheerfully tolerated a never-ending amount of inconvenience. It is doubtful that this effort would have been accomplished without her continued thoughtfulness.

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CHAPTER I

INTRODUCTION

A. Purpose and Scope of Work

A monotube forced-circulation concept in which preheating, evaporation, and superheating is a continuous process in either single or parallel tubes offers promise in the marine propulsion field. Such "once-through" boilers possess most of the advantages of forced-circulation boilers with the added attraction of a steam generator in which the order of magnitude of the thermal inertia is equal to that of the mechanical energy conversion machinery. Although many types of this class of steam generator have been recently installed in some large power stations, there is no real understanding of the transient response within the two-phase flow area of the tubes. Profos [1] has presented a combined graphical and mathematical method by which the transient characteristics can be determined from the design data of the steam generator.

The dynamic control behavior of forced-flow evaporator systems is of considerable importance in the control of once-through forced-circulation boilers. It, likewise, plays a role in drum-type boilers with steaming economizers (drum-type boilers without economizers are designed under nucleate boiling conditions). Dynamic behavior decides the character of the feed control system. It influences the properties of the boiler as a whole, and as such can be used as a vital tool in boiler design.

A knowledge of the dynamic behavior of forced-ciculation systems under transient conditions would permit the development of a transfer function for use on the analog computer (Appendix A). By employing control synthesis much could be learned concerning the response of such systems without building and testing large scale prototypes. The engineer could predict the changes in boiler response that would be brought about by varying the basic parameters of the generating system.

During operation evaporator systems are subject to various disturbances in the form of input variables. Changes in feedwater flow, W, and heat flow of the evaporator surface, q, influence both the displacement of the commencement and the end point of evaporation. This results in changes in the size of the heating surface of the preheater (economizer), evaporator, and superheater. To predict these changes in the commencement and end point of evaporation (hereafter referred to as the saturated interface and superheat interface, respectively) an experimental study of the two-phase region dynamics is required. The drastic assumptions required in any analytical solution of the partial differential equations involved limit their ability in fully describing the boiling phenomenon under both steady state and transient conditions. Usually, perturbation theory centered on several selected operating points must be employed.

Experimental work done by Dengler [2], Kozlov [3], and Armand [4] describes boiling phenomenon in both natural-circulation and forced-circulation units. Bergles and Rohsenow [5] studied forced convection boiling heat transfer and tube burnout

both experimentally and analytically. Wallis and Heasley [6] and Stenning [7] analytically treat two phase oscillations and instabilities. All of this work, however, is concerned with steady state conditions. No published work other than that of Profos [1] could be found which treats the transient two-phase flow problem.

It is the purpose of this thesis to experimentally determine interface movement under transient conditions.

B. Discussion of Boiling Mechanism in Tubes

Dengler [2] discusses five types of two-phase vertical flow under forced circulation conditions: bubble, slug, slug-annular, turbulent-annular and smooth annular. Slug-annular and smooth annular dominate the range of vaporization of most practical interest. He related the various flow phenomena to fraction vaporization, pressure, and mass flow rate. The type of flow pattern had little effect on the heat transfer coefficient over the initial range of vaporization. Following Dengler the regimes are described:

a. Bubble Flow

Bubble flow is observed at extremely low weight per cent vaporization. The bubbles are at first small and well distributed throughout the liquid phase. As vaporization increases the bubbles grow in size, decrease in numbers, and occupy the center of the tube. Bubble flow persists only over the initial stages of vaporization. It disappears when less than 0.2 per cent of the liquid is vaporized.

b. Slug

With the vaporization of more liquid, the bubbles coalesce and form slugs of vapor which alternate with liquid along the tube. The slugs then unite and displace liquid from the center of the tube.

c. Slug-Annular

The slug-annular flow regime is typified by a sector of extreme agitation. Annuli of liquid, temporarily held up on the wall by the growing slugs of vapor, repeatedly collapse and fall back into the center of the tube. Heavy rings of liquid travel up the tube wall and then tumble back down to temporarily block the vapor. Pressure builds up, and the vapor explodes through the fallen slug of liquid again forcing it up the tube. The pressure falls and the cycle is again repeated. Slug-annular flow is thus marked by severe pressure drop fluctuations. At low flow rates slug-annular flow lasts until a substantial portion of the liquid, 10 to 20 per cent, has been vaporized.

d. Turbulent-Annular

In this regime, the agitation subsides and the vapor, now moving at a high velocity, forces the liquid up the tube wall. The central core of vapor is surrounded by a stable annulus of rapidly moving liquid. The violent fluctuation of slug-annular flow is replaced by a fine grained turbulence confined to the area of the tube wall. Pressure fluctuations are greatly reduced and stability approaching that of one-phase flow is achieved.

e. Smooth-Annular

With still a greater weight per cent of vaporization the flow at the tube wall becomes smoother, and apparently the thickness of the wall film grows thinner. The transition between the two annular types takes place suddenly. In smooth-annular flow it is quite possible that part of the liquid has left the tube wall and entered the central core as spray.

Kozlov [3] recognizes six basic types of flow: bubble flow, plug flow, plug-dispersion flow, emulsion flow, film-emulsion flow, and drop flow. This description compares favorably with that of Dengler. It differs in the respect that Kozlov can differentiate two regimes, emulsion flow and film-emulsion flow, in that area which Dengler describes as Turbulent-Annular.

The author's observations during the experimental work of the thesis justifies the acceptance of Dengler's definitions. These are more visually descriptive of the occurrence within the two phase region and can be detected quite easily.

The range of vaporization during which bubble and slug flow exists is extremely narrow. Dengler's data [2] indicates that both disappear when less than 0.2 per cent of the liquid has evaporated. He concludes that the effect of these mechanisms, if any, on evaporation of liquids in vertical tubes may be ignored, since the local heat transfer coefficient changes very little within these regimes.

Buchberg, et al. [8] conducted an investigation of conditions required for the inception of boiling. Axial wall-

temperature variation and wall temperature fluctuation were observed to determine the incipient boiling point. The difference in temperature between a downstream location and a location near the inlet was plotted versus heat flux for constant pressure and flow. A sharp peak in the curve defined the heat flux and the point of incipient boiling.

The data of Swenson, et al. [9] indicates that wall temperature peaks at a steam quality greater than 0. They indicate that a transition has been made at this peak from nucleate boiling to film boiling. Becker, et al. [10] in graphical form present the relation of pressure, temperature (outside wall, T_{wo}, inside wall, T_{wi}, and bulk temperature, T_b) and steam quality distribution along uniformly heated test sections. This data shows that T_{wo}, T_{wi}, and T_b all peak within a 2 - 3 inch length of a 3120 mm heated vertical tube. It is significant to note that the peak T_{wo} corresponds to 0 quality steam.

Bergles and Rohsenow [5] believe that the peak wall temperature indicates the start of fully-developed boiling rather than the incipient boiling point. They further feel that it is not necessary to determine the point of incipient boiling at high pressures, as it is separated from fully-developed boiling by only a few degrees of wall superheat.

It can, therefore, be concluded that there is a direct correlation in T_{wo}, T_{wi}, T_b, and the location of the commencement of boiling. This correlation is the key to the prediction of interface movement under transient conditions.

C. Apparatus and Equipment

Two distinct phases of study with their accompanying installations were employed to obtain the required background in two-phase flow for the writing of this thesis.

1.) First Installation

The first installation was constructed at the U.S. Naval Boiler and Turbine Laboratory, Naval Shipyard, Philadelphia, Pennsylvania. With laboratory assistance a fifteen foot monotube boiler unit was erected. The unit was equipped with the required pumps, tanks, demineralizers, etc. that enabled it to operate as a steam generator (FIG. 1). It was instrumented with thermocouples that extended into the flow regime to trace the variance of bulk temperature, T_b, with space and time. The erection of such a test unit brought up subsidiary problems. A simple yet effective constant refererence temperature junction for use with one hundred and eighteen thermocouples was developed. A means for obtaining ultrapure distilled water was solved. A monitoring system for obtaining the time variation in temperature of the one hundred and eighteen installed thermocouples was developed (FIG. 2). A platinum resistance thermometer for the measurement of average temperature was developed as an outgrowth of this program (FIG. 3). Additional views of the installation are presented in FIG. 4 and FIG. 5. Mechanical problems caused a premature failure of the unit before the compilation of the required data. The program is still an active one at the laboratory and a new design is being contemplated.

FIGURE 1
SCHEMATIC VERTICAL TUBE
EVAPORATOR TEST INSTALLATION
NBTL

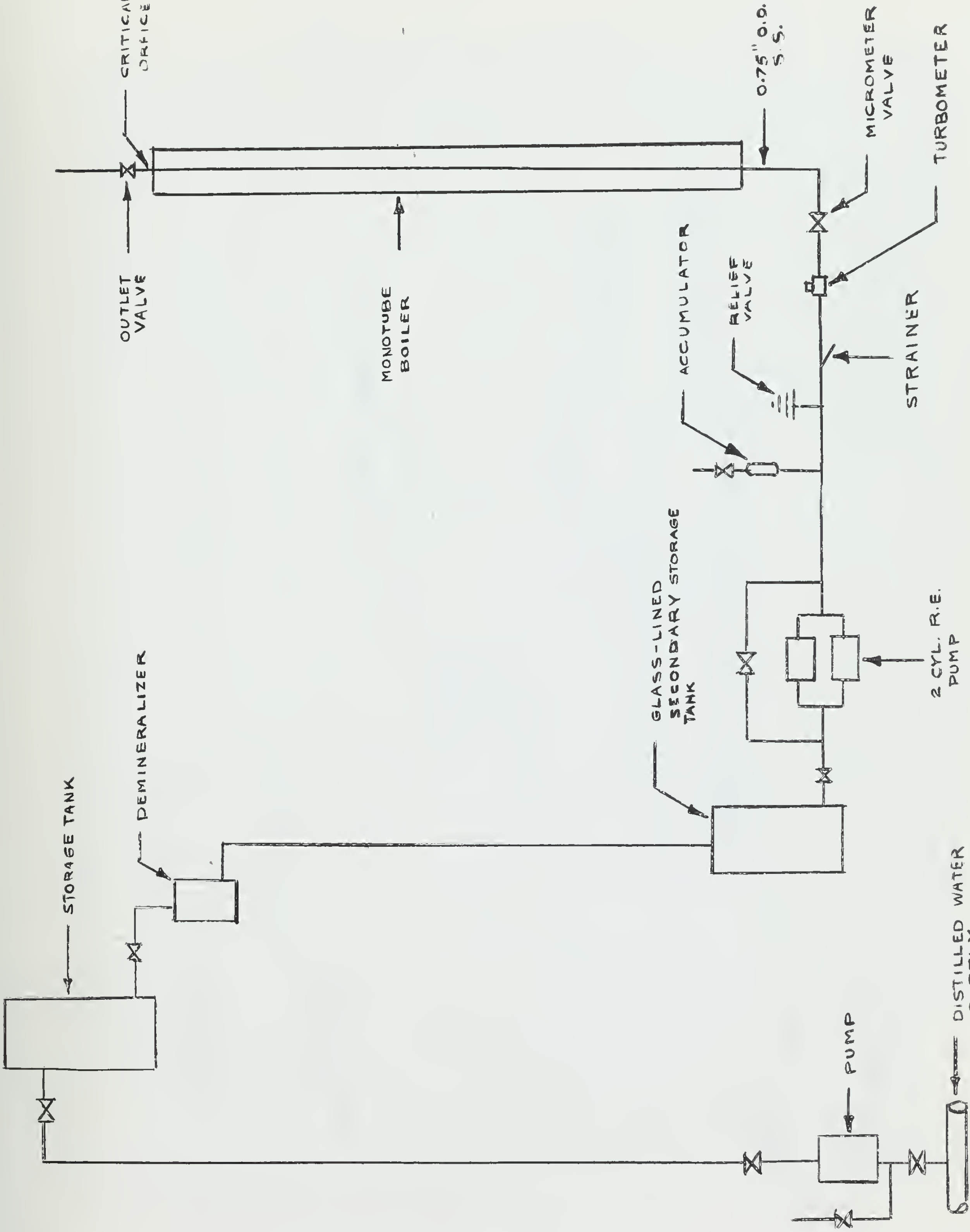
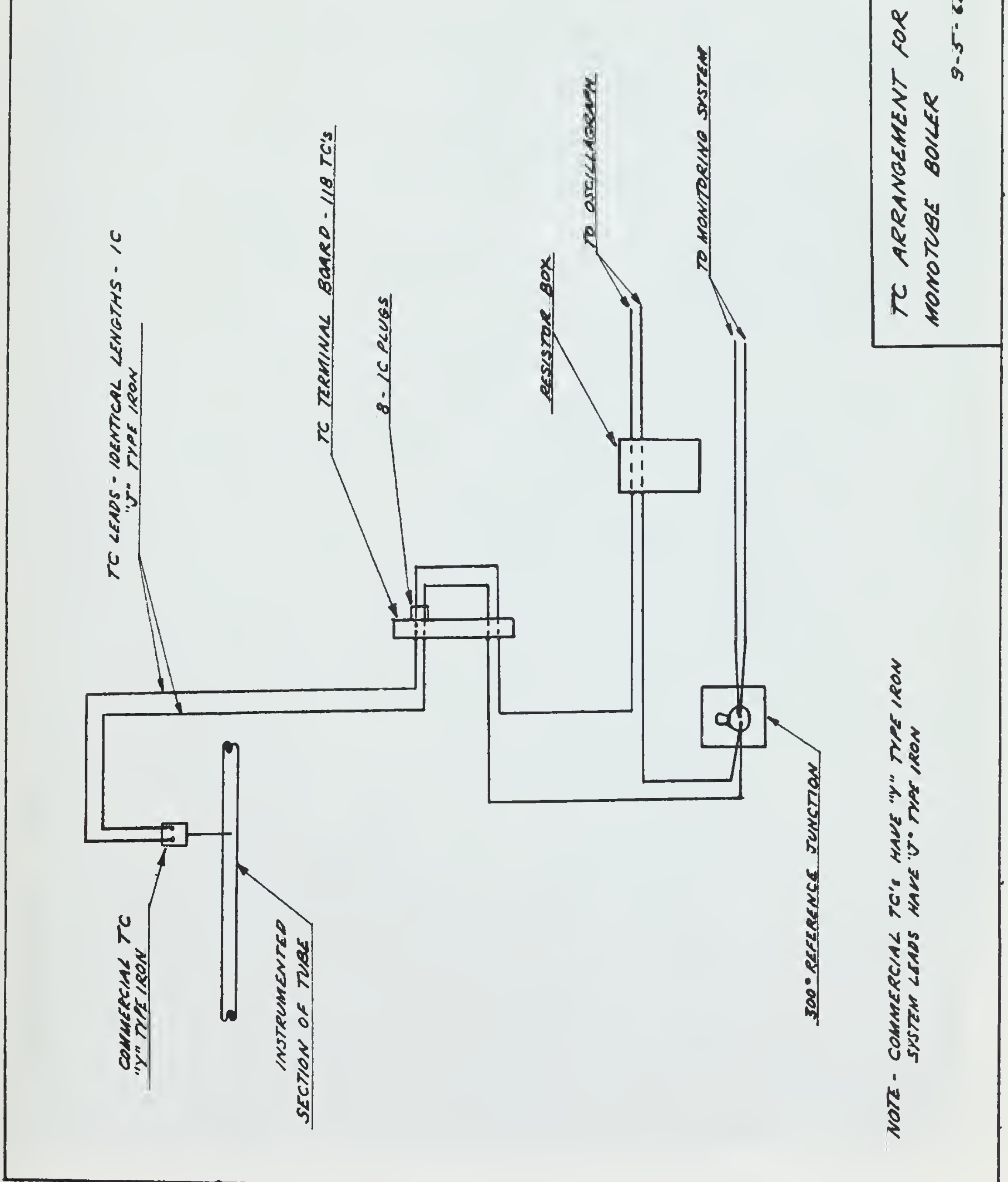


FIGURE 2


TC LEADS - IDENTICAL LENGTHS - IC
"J" TYPE IRON
TC TERMINAL BOARD - 118 TC's
8 - IC PLUGS
RESISTOR BOX
TO OSCILLAGRAPH
TO MONITORING SYSTEM
TC ARRANGEMENT FOR
MONOTUBE BOILER
COMMERCIAL TC
"Y" TYPE IRON
INSTRUMENTED
SECTION OF TUBE
300° REFERENCE JUNCTION
NOTE - COMMERCIAL TC's HAVE "Y" TYPE IRON
SYSTEM LEADS HAVE "J" TYPE IRON

(U.S. Navy Photo)

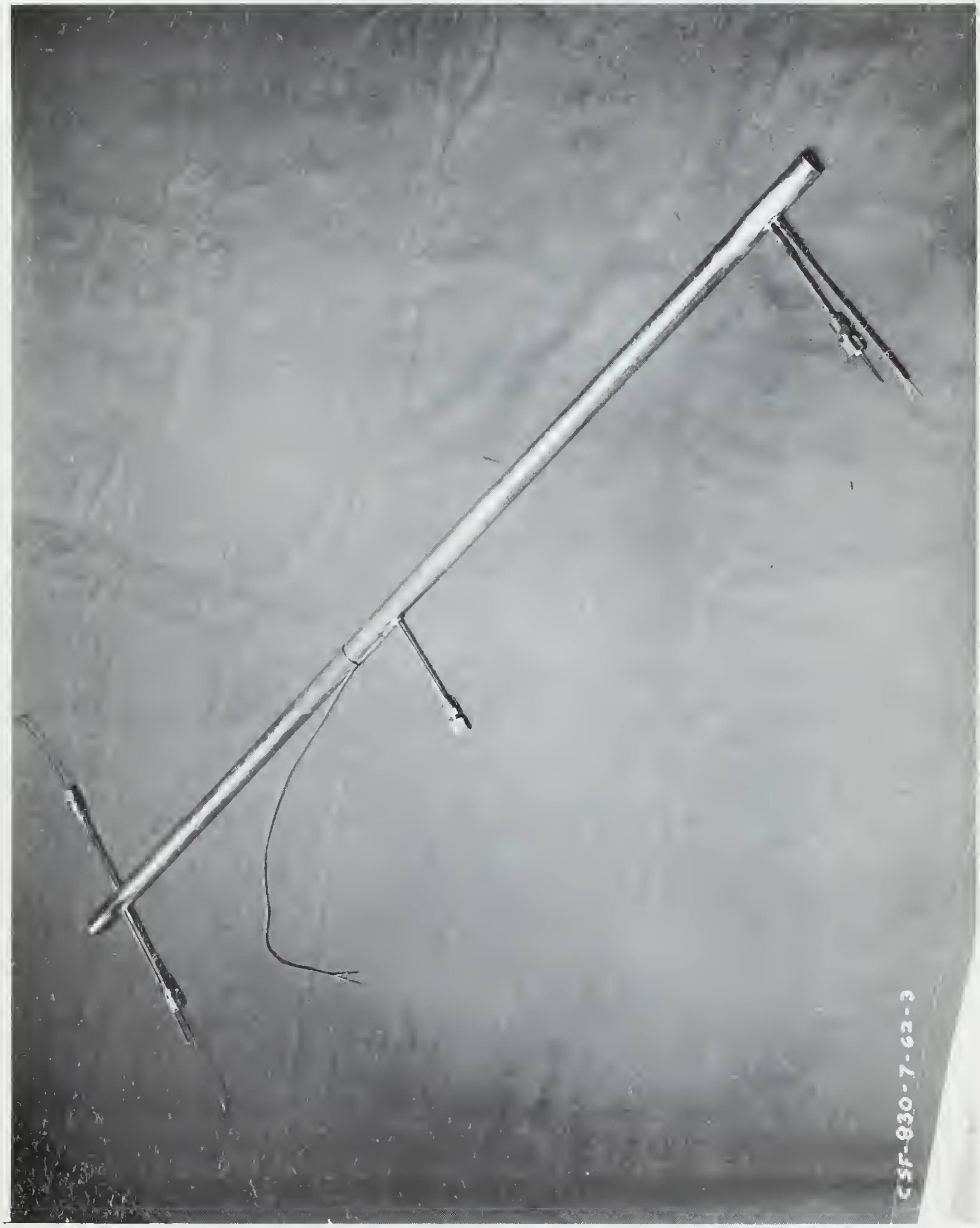


FIGURE 3

PLATINUM RESISTANCE THERMOMETER FOR THE MEASUREMENT OF AVERAGE TEMPERATURE IN A REGION OF TWO-PHASE FLOW.

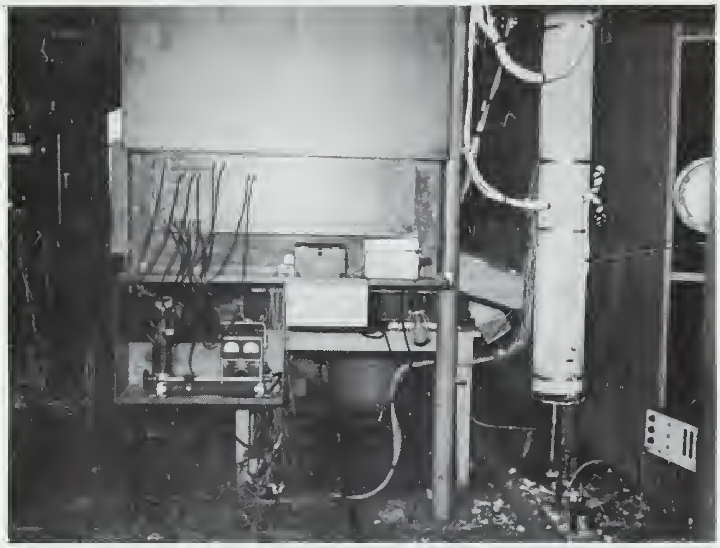


FIGURE 4
VERTICAL TUBE EVAPORATOR TEST INSTALLATION, NBTL
(From left: Constant Temperature Reference, TC Terminal Board, and Monotube Boiler)

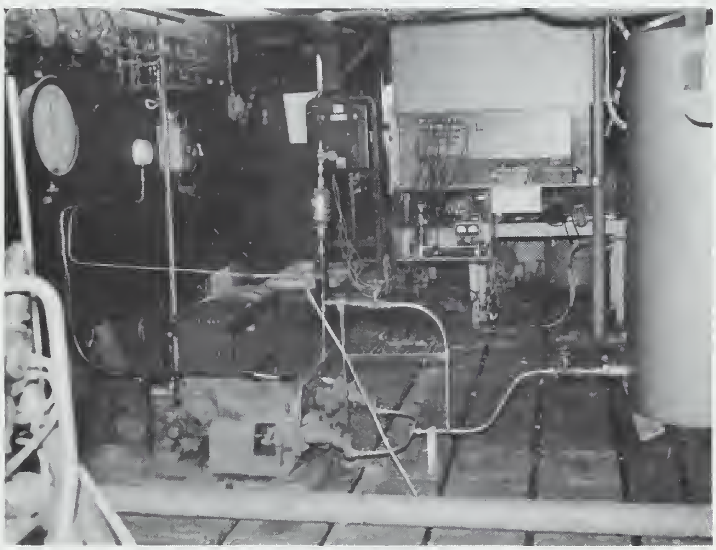


FIGURE 5
VERTICAL TUBE EVAPORATOR TEST INSTALLATION, NBTL
(Visible left foreground 2 cyl. R.E. Pump with accumulator; Right, Glass-lined Secondary Storage Tank)

2.) Second Installation

The second phase of the program was conducted at the Experimental Projects Laboratory, M.I.T. The author concluded that any attempt to measure bulk temperature, T_b, within a tube by means of thermocouples must be discarded. An immense mechanical-joining problem exists in the areas where the thermocouples pierce the generating tube, since the tube must necessarily be heated by some form of electrical device to obtain a uniform heat distribution. In addition, the flow disturbance resulting from the protrusion of the thermocouples into the tube gives data which can not be correlated to the geometry of boiler tube flow.

In the second installation the use of ultrapure distilled water was discarded. This experimenter felt that in using such extreme refinements, conditions were being produced that were too laboratory directed for extrapolation to general industrial equipment. Carried to the extreme, within a liquid containing no colloidal impurities, gaseous or solid, vaporization would be impossible. Lastly, simplicity in design was a critical factor consideration.

a.) General Description

The equipment consisted of the following components:

(1.) Vertical Evaporator

(2.) Visual Observation Section

(3.) Condenser

(4.) Electrical Circuit

(5.) Flow System

(6.) Thermocouple Measuring System

A Schematic of the installation is shown in FIG. 6.

b.) Detailed Description

(1.) Vertical Evaporator

The vertical evaporator was constructed from a ten foot section of Type No. 304 (18-8), stainless steel tubing, O.D. 0.375", wall thickness 0.028". To obtain a reasonable electrical power input for steam generation the two electrical heaters were installed in parallel. Each heater consisted of thirteen pyrex glass tubes 4 feet in length and 4 mm in diameter (standard type, wall thickness 0.8 mm). The pyrex tubing was firmly secured about the outside circumference of the stainless steel tubing forming an annular ring. In each section, through the centers of the pyrex glass tubes in the longitudinal direction, there was woven a continuous length of size 18 Chrome "A" wire with a current rating of 23 amps, 0.412 Ohms/Foot resistance rating (FIG. 7). The heat generating area was lagged with standard mineral wool piping insulation (sectional length 3 ft.). Power was supplied to the unit from a 230 volt D.C. bus. Evaporator operating pressure was 20 psia.

(2.) Visual Observation Section

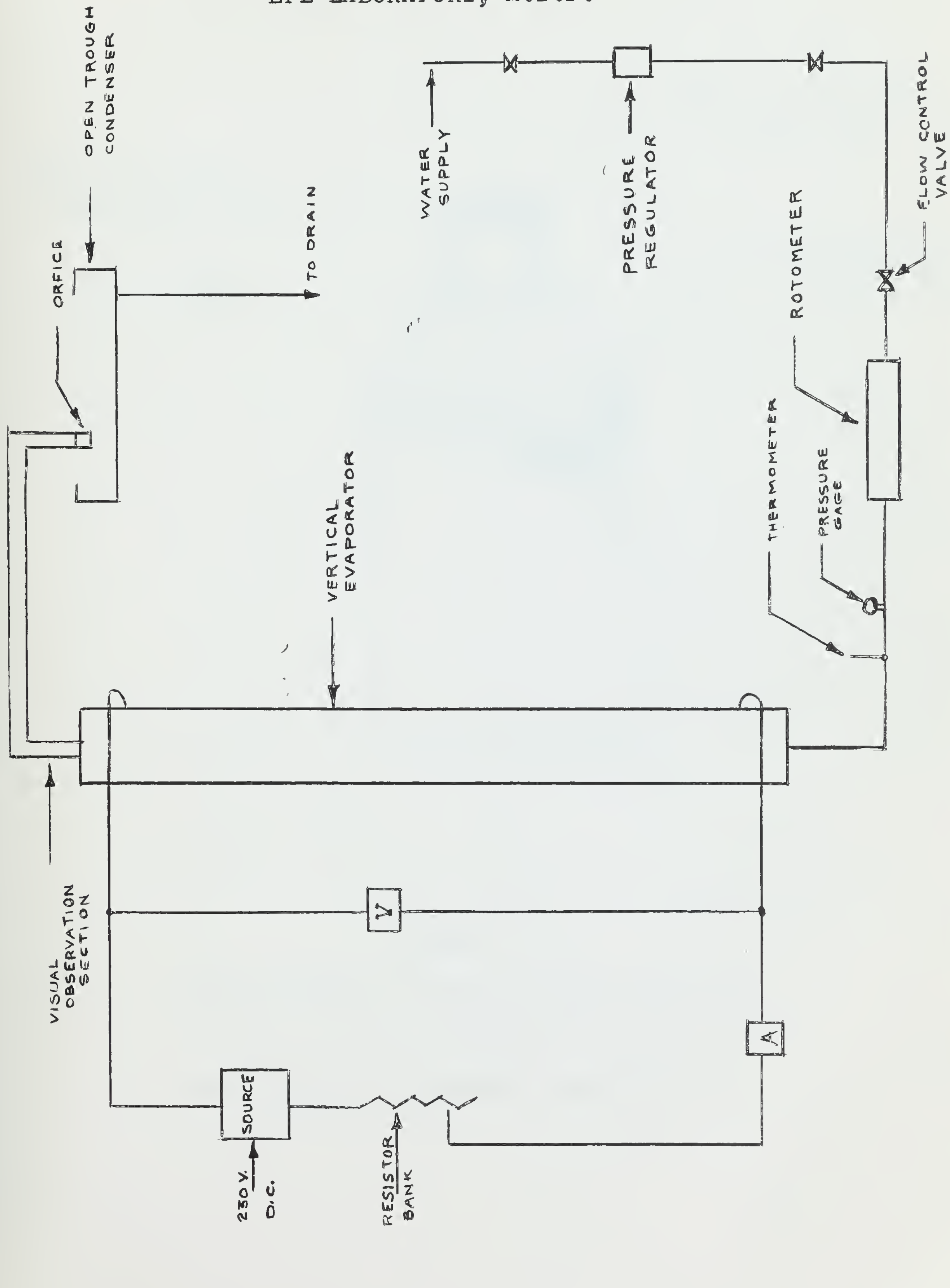
Secured to the top of the evaporator, at the end of the uppermost heating unit, was the visual observation section. This section consisted of a 2 ft. length of standard Pyrex Tubing, O.D. 3/8", wall thickness 5/64". Discharge piping to the open condenser was attached to this section. Back pressure was built up within the evaporator by the installation of an orfice at the end of this discharge piping. The visual

FIGURE 6

SCHEMATIC VERTICAL TUBE

EVAPORATOR TEST INSTALLATION

EPL LABORATORY, M.I.T.



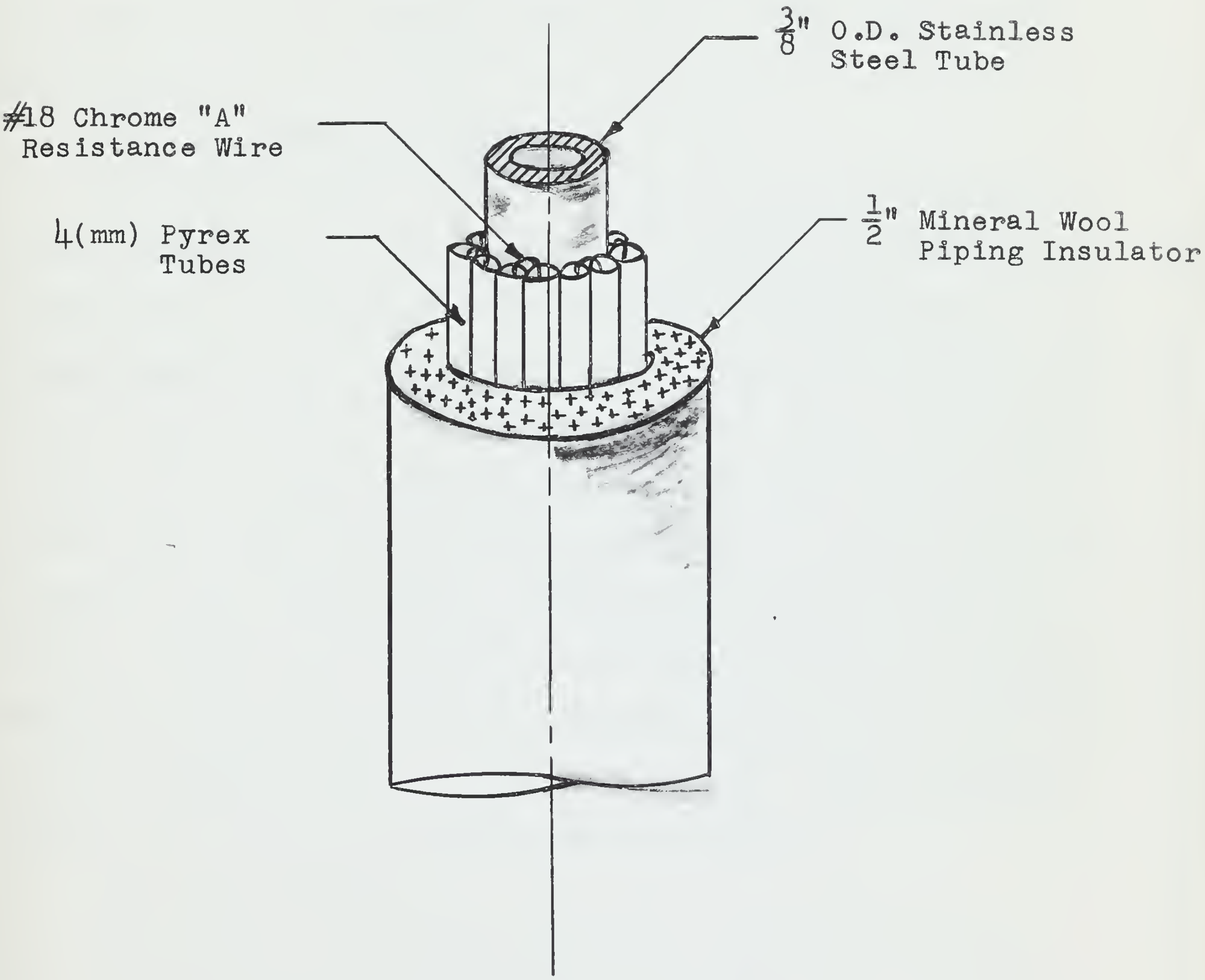


FIGURE 7

OBLIQUE SKETCH VERTICAL TUBE EVAPORATOR

(A portion of the insulation is removed to show interior evaporator construction)

observation apparatus permitted observation of the two-phase flow phenomena.

(3.) Condenser

The condensing unit was an open trough type condenser. Discharge from the condenser was led to a fresh water drain. The steam cycle was operated as an open cycle.

(4.) Electrical Circuit

The steam within the vertical tube evaporator was generated by electricity. Electricity was adopted because of the simplicity of inducing the required step changes in heat flow rate, q, during the experimental runs. Power was supplied from a 230 Volt D.C. bus. Step changes in heat flow rate were accomplished through a variable resistor (8 pole switch stove rated at 40 amps, 230 Volt D.C.) connected in series with the parallel Chrome "A" heaters. Current input was measured by means of a Weston D.C. Ammeter (0-40 amps rating). Voltage across the generating unit was measured by means of a Weston D.C. Voltmeter (3 scale, 0-300 volt rating). The electrical circuit is illustrated schematically in FIG. 6.

(5.) Flow System

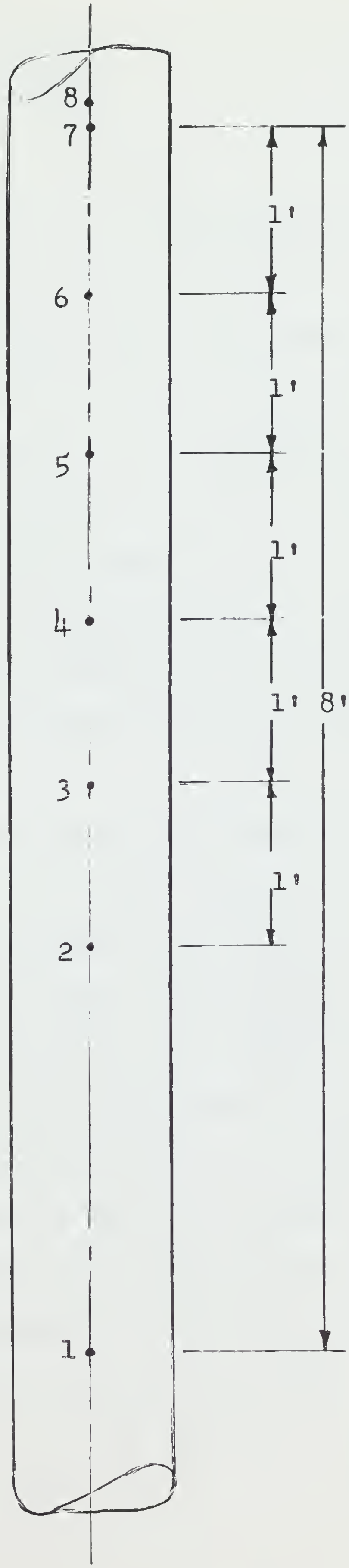
The system of flow was as follows. Water was received from Laboratory Fresh Water Piping (60 psi) and passed through a Mason-Neilan Regulator, Type 33-1, rating 200 psi, range 2 psi to 20 psi. This regulator isolated the feed water system from any exterior water pressure variation. The total weight flow rate, W, was measured by a Fischer-Porter Rotometer (Scale 0-20). Flow rate was controlled through a 3/8" needle

valve and input pressure was measured by means of a standard pressure gauge (0-30 psi). After undergoing boiling in the test section, the steam passed through the visual observation section. Vapor was condensed in the open trough condenser.

<u>(6.) Thermocouple Measuring System</u>

Eight thermocouples were secured to the outside wall of the generating tube. Thermocouple placement is illustrated in FIG. 8. Seven thermocouples and their leads were made of copper-constantan. These were placed along the length of the generating tube and were used to measure steady state temperature variation. The thermocouple leads were attached to a selector switch. This switching arrangement permitted individual monitoring of each thermocouple on a Leeds and Northrup Millivolt Potentiometer. The eigth thermocouple, iron-constantan, was secured to the wall at the uppermost heater outlet. It was used to measure wall temperature variation under transient conditions and was monitored individually by the Leeds and Northrup Millivolt Potentiometer.

FIGURE 8
GENERATING SECTION
THERMOCOUPLE PLACEMENT



CHAPTER II

PROCEDURE

A. Experimental Procedure

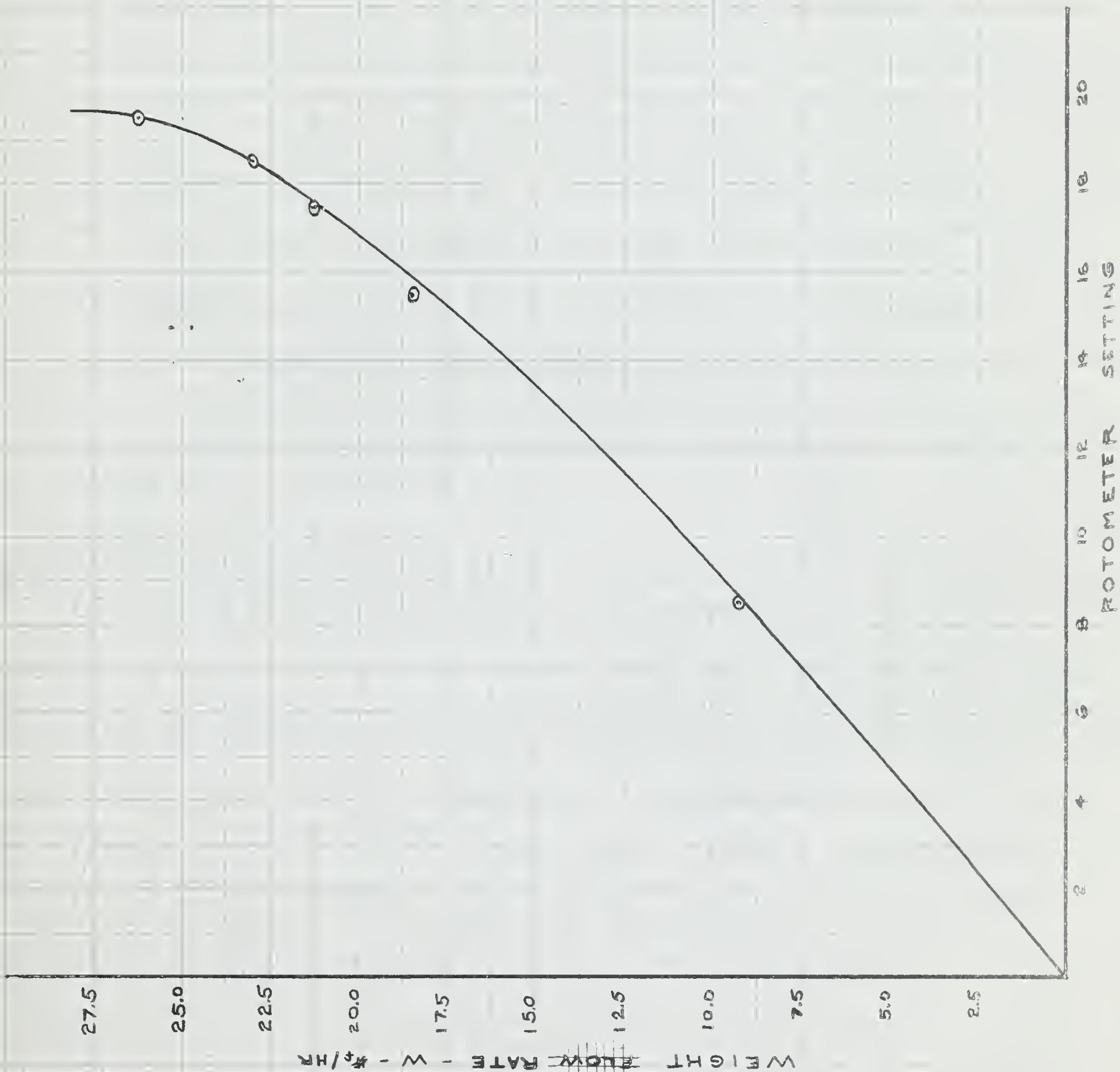
Feedwater was introduced into the apparatus through the rotometer and needle control valve. A curve of rotometer reading versus weight flow rate had been prepared in a series of calibration runs (Calibration Curve, FIG. 9). The weight flow rate, W, was set. The electrical heating elements were energized to obtain a given heat flow rate, q, and the system was brought up to steady state conditions. At the end of this process, which generally took approximately one hour, outside wall temperature, T_{wo}, was monitored on Thermocouple 8 to insure that a steady state condition did exist (indicated by no rise in T_{wo}).

When steady state conditions were established, wall temperatures on Thermocouples 1 - 7 were recorded. The flow regime existing in the visual observation section was recorded. Any one of the four types of investigation runs was then commenced by the inducement of either step changes in flow rate, W, or heat flow rate, q. Wall temperature variation was measured on Thermocouple 8 at 15 second intervals. The existing flow regime in the visual observation section was also recorded at these times. Upon reaching a new steady state condition, wall temperatures, on Thermocouples 1 - 7 were recorded along with the final steady state flow regime. It required approximately

FIGURE 9

ROTOMETER CALIBRATION

CURVE



8 - 15 minutes to reach a new steady state condition in the water and heat flow rates in which investigations were conducted.

The types of runs on which studies were undertaken were as follows:

1.) System maintained at a constant weight flow rate, but heat flow rate increased (step change).

2.) System maintained at a constant weight flow rate, but heat flow rate decreased (step change).

3.) System maintained at a constant heat flow rate, but weight flow rate increased (step change).

4.) System maintained at a constant heat flow rate, but weight flow rate decreased (step change).

A sample data sheet (TABLE IV) and work sheet (TABLE V) are illustrated in Appendix C.

B. Analytical Procedure

In the analytical procedure, the thermocouple readings were converted to wall temperature readings. At any given time after the step change, the wall temperature, T_{wo}, on Thermocouple 8 was subtracted from the initial wall temperature. The resulting ΔT values were plotted versus time. Visual observations were then correlated with time and temperature. A curve was fitted to these experimental points. The curve of

$$\Delta T = \Delta T_{\infty} f(t) \tag{1}$$

was determined. ΔT_{∞} is defined as the difference in the two steady state temperature readings (commencement of run and termination of run).

$$\Delta T_{\infty} = T_{\infty} - T_1 \tag{2}$$

Since the positions of Thermocouples 1 - 7 were known along the axial length of the tube, FIG. 8, it was quite simple to locate the point of the commencement of evaporation by plotting wall temperature, T_{wo}, versus thermocouple location curve. The peak in the T_{wo} curve corresponded to the location of the commencement of evaporation. The heat flow rate, q, and the weight flow, W, were so controlled in each run that at either the commencement of the run or after the step change in the system variables, the commencement of nucleation was just occurring at the end of the generating section. In other words, at one level a peak in T_{wo} (read on Thermocouples 1 - 7) was apparent and its position along the length of the tube could be ascertained. At the other level, the T_{wo} at each location showed a steady rise which gave an indication of the heating of a subcooled liquid to a point of the commencement of first boiling. For any run the length of the generating tube minus the distance down the tube at which the peak occurred was equal to the total length of saturated interface movement between two steady state operating levels, ΔL_{∞}.

$$\Delta L_{\infty} = L - L_p \qquad (3)$$

Equation 1 was then related to ΔL_{∞} and a curve

$$\Delta L = \Delta L_{\infty}\, f(t) \qquad (4)$$

was obtained. It was assumed that the f(t) for both temperature rise, ΔT, and interface movement, ΔL, was the same. This is justifiable from the fact that a general correlation giving critical wall superheat as a function of pressure and heat flux can be readily developed to predict the point of incipient

boiling [5]. For a given system there is a critical wall superheat at which nucleation will commence. As system variables (heat flow rate and weight flow rate) are changed the point of incipient boiling will move in the tube. The given critical temperature will move either up or down the length of the generating section. This critical temperature movement is exactly the movement of the saturated interface. If in an experimental run a given critical temperature exists at the outlet of the generating section and if nucleation bubbles are apparent, then the time variation of temperature on Thermocouple 8, after a change in system variables, is the same f(t) as the critical temperature movement with time, except that it is opposite in direction. For instance, if T_{wo} at Thermocouple 8 rose to a new higher value (observation of the boiling phenomena in the visual observation section would show a change from nucleation bubbles to slug-flow, and thence to slug-annular flow), the point of incipient boiling would be moving downward into the evaporator.

For superheat interface movement, outlet temperature variation with time was correlated with visual observation for a trend in the superheat interface movement. The wall temperature peaks have no value, however, in locating the end point of evaporation. ΔL_{∞} can not be obtained. More emphasis must be placed upon visual observation.

CHAPTER III

RESULTS

In the course of this study, fifty-three various types of runs were conducted. Nine representative ones have been presented here. A sample calculation for Run 3-1 is given in TABLE VI, Appendix C.

The results are presented in graphical form (FIGURES 10-38). For each run (except 101c), three graphs are given representing:

1. $\Delta T = \Delta T_\infty f(t)$
2. $\Delta T_p = f(L)$
3. $\Delta L = \Delta L_\infty f(t)$

The first type includes indications of the visual observations that were conducted. The second graph in each series, $\Delta T_p = f(L)$, gives the total interface movement between two steady state levels for a particular run. The last curve is the derived curve of interface movement. A summary of original data for all curves is presented in tabular form in Appendix D (TABLES VII and VIII). Calculated data for the runs is also included (TABLES II and III, Appendix B).

Composite plots of saturated interface movement for W constant (FIG. 36) and q constant (FIG. 37) are also presented. A summary of ΔL_∞ for the various runs can be seen in TABLE I. Time constant, "a", variation with W for step changes in q is shown in FIG. 38.

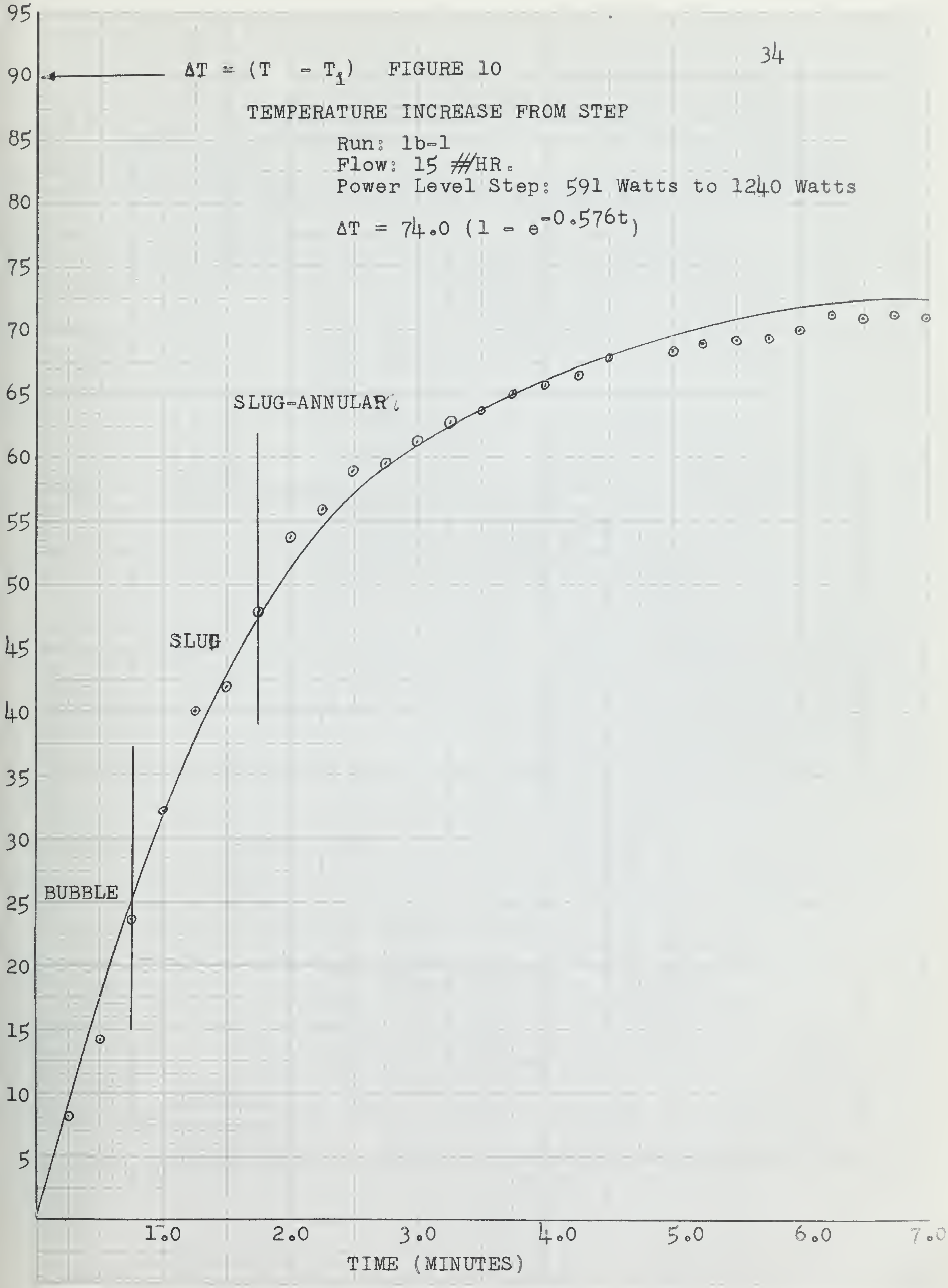


FIGURE 10

TEMPERATURE INCREASE FROM STEP

FIGURE 11

LOCATION PEAK WALL TEMPERATURE

Run: 1b-1
Flow: 15 #/HR.
Power Level Step: 591 Watts to 1240 Watts
ΔL_{∞} = -2.65 Ft.
Scale: 1" = 1'
— — — = Conditions Before Step

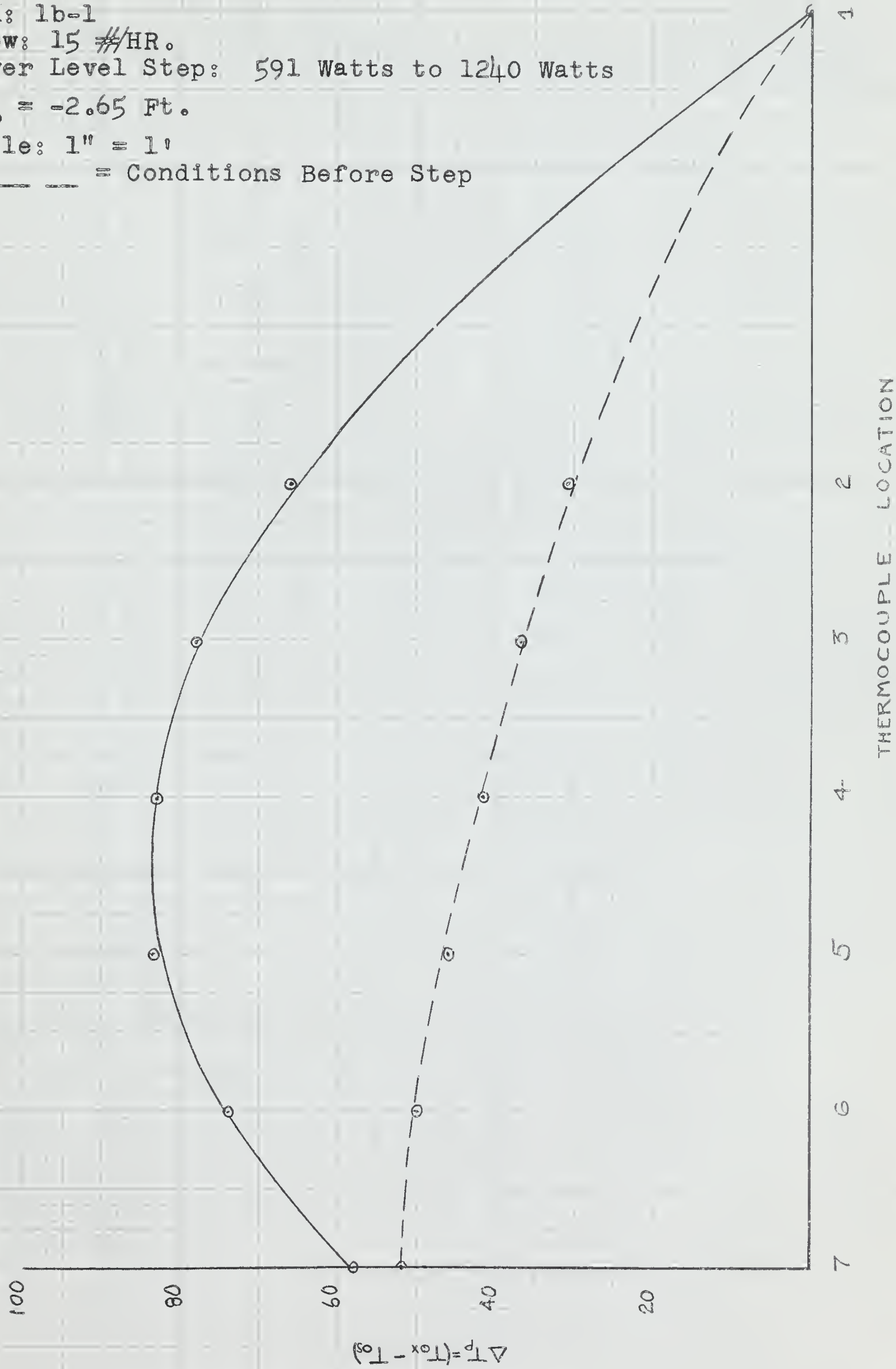
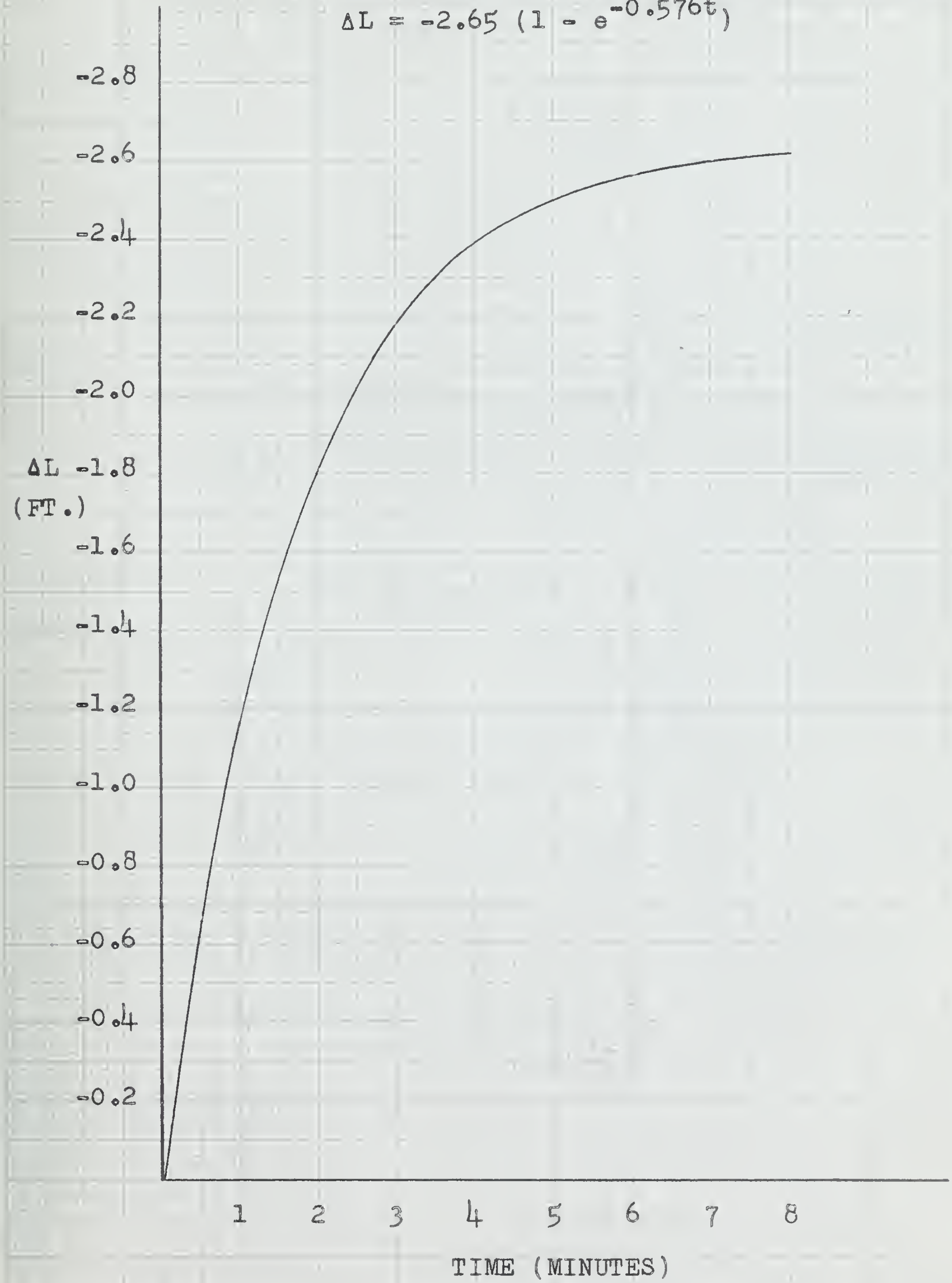


FIGURE 12

TOTAL DERIVED INTERFACE MOVEMENT

Run: 1b-1
Flow: 15 #/HR.
Power Level Step: 591 Watts to 1240 Watts
$\Delta L = -2.65\,(1 - e^{-0.576t})$



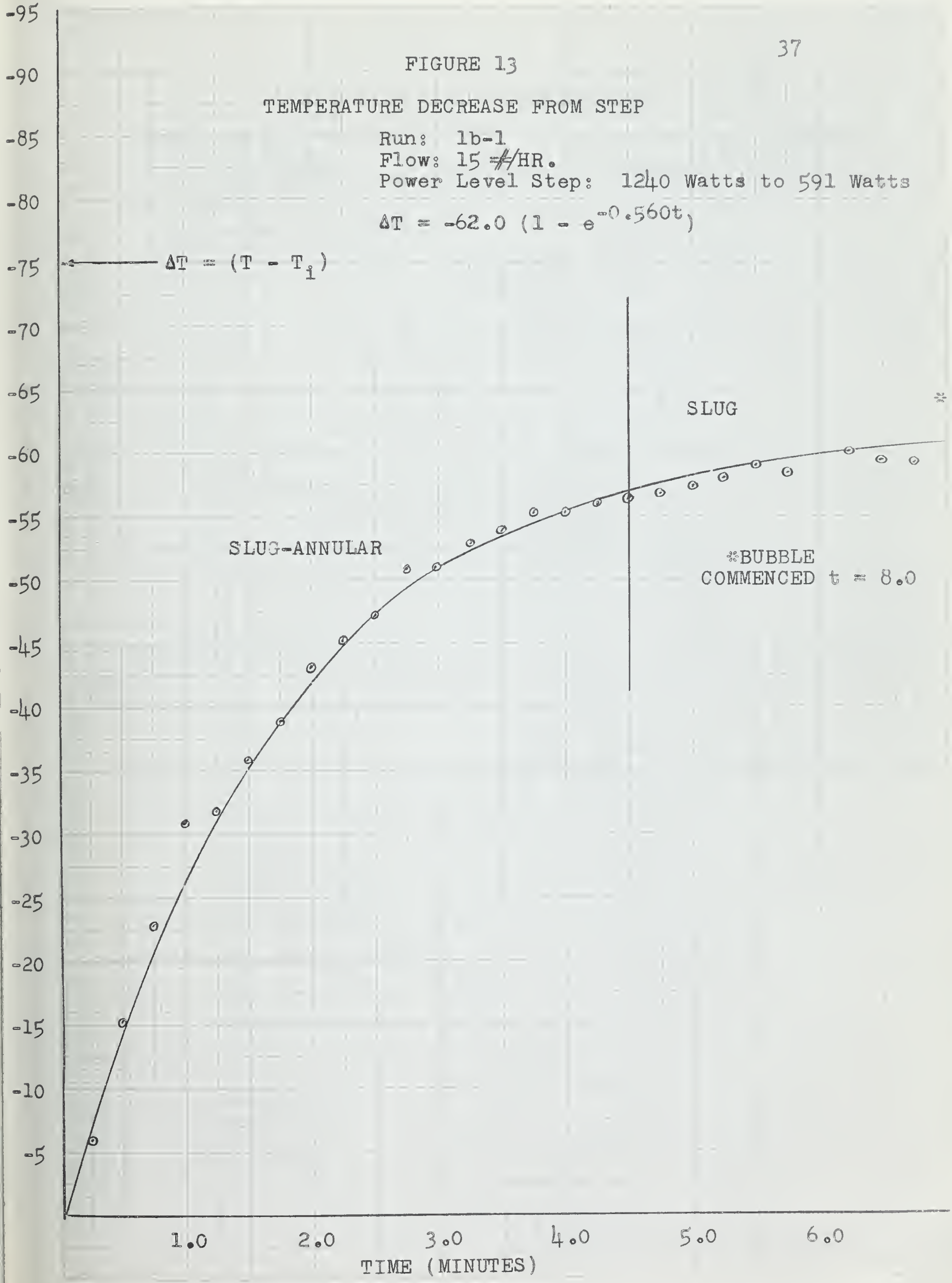


FIGURE 13
TEMPERATURE DECREASE FROM STEP
Run: 1b-1
Flow: 15 #/HR.
Power Level Step: 1240 Watts to 591 Watts
$\Delta T = -62.0 (1 - e^{-0.560t})$
$\Delta T = (T - T_i)$
SLUG
SLUG-ANNULAR
*BUBBLE COMMENCED t = 8.0
-95
-90
-85
-80
-75
-70
-65
-60
-55
-50
-45
-40
-35
-30
-25
-20
-15
-10
-5
1.0
2.0
3.0
4.0
5.0
6.0
TIME (MINUTES)

FIGURE 14

LOCATION PEAK WALL TEMPERATURE

Run: 1b-1
Flow: 15 #/HR.
Power Level Step: 1240 Watts to 591 Watts
ΔL_{∞} = 2.6 FT.
Scale: 1" = 1'
__ __ __ = Conditions Before Step

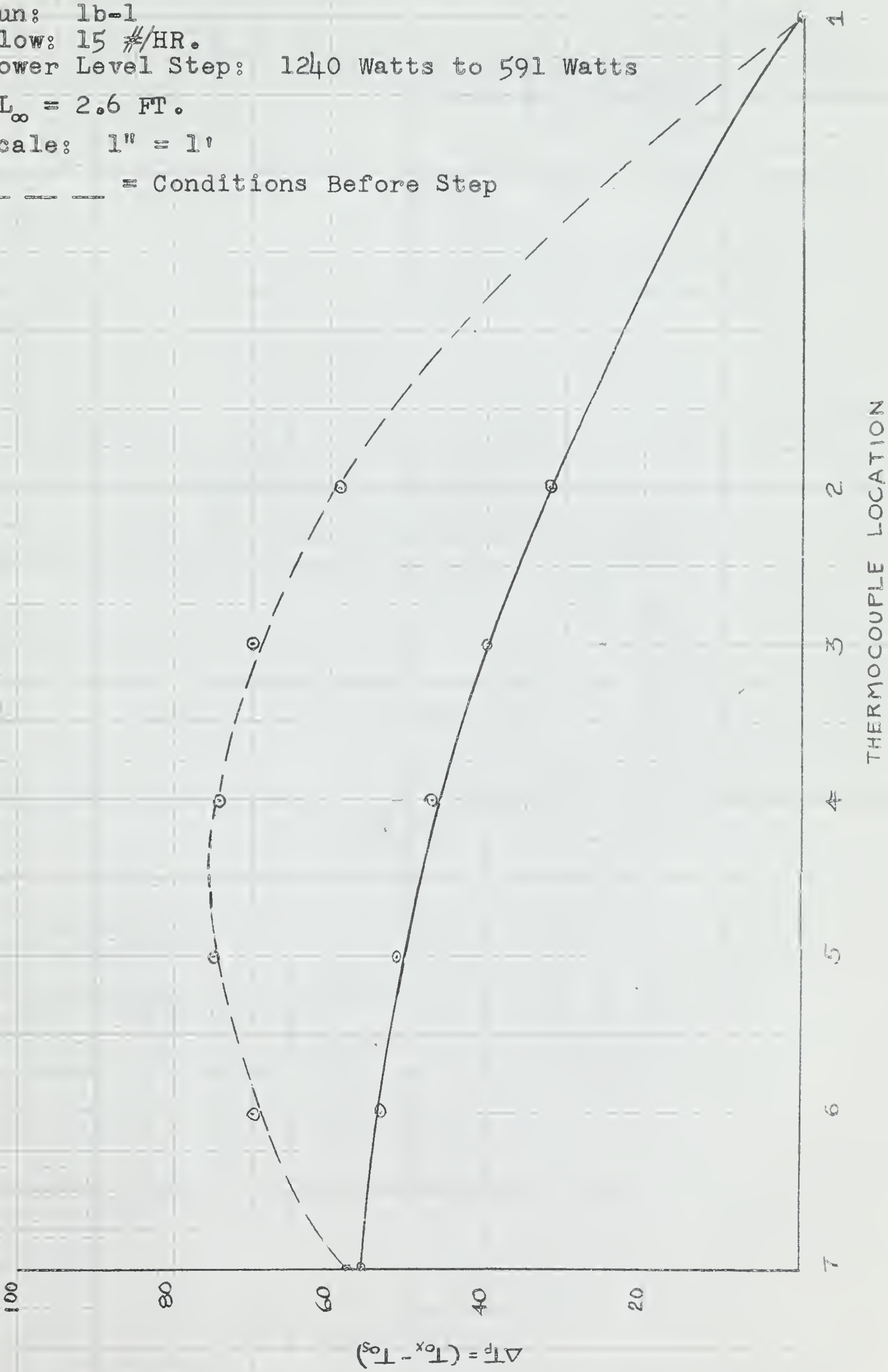


FIGURE 15

TOTAL DERIVED INTERFACE MOVEMENT

Run: 1b-1
Flow: 15 #/HR.
Power Level Step: 1240 Watts to 591 Watts
$\Delta L = 2.6\,(1 - e^{-0.560t})$

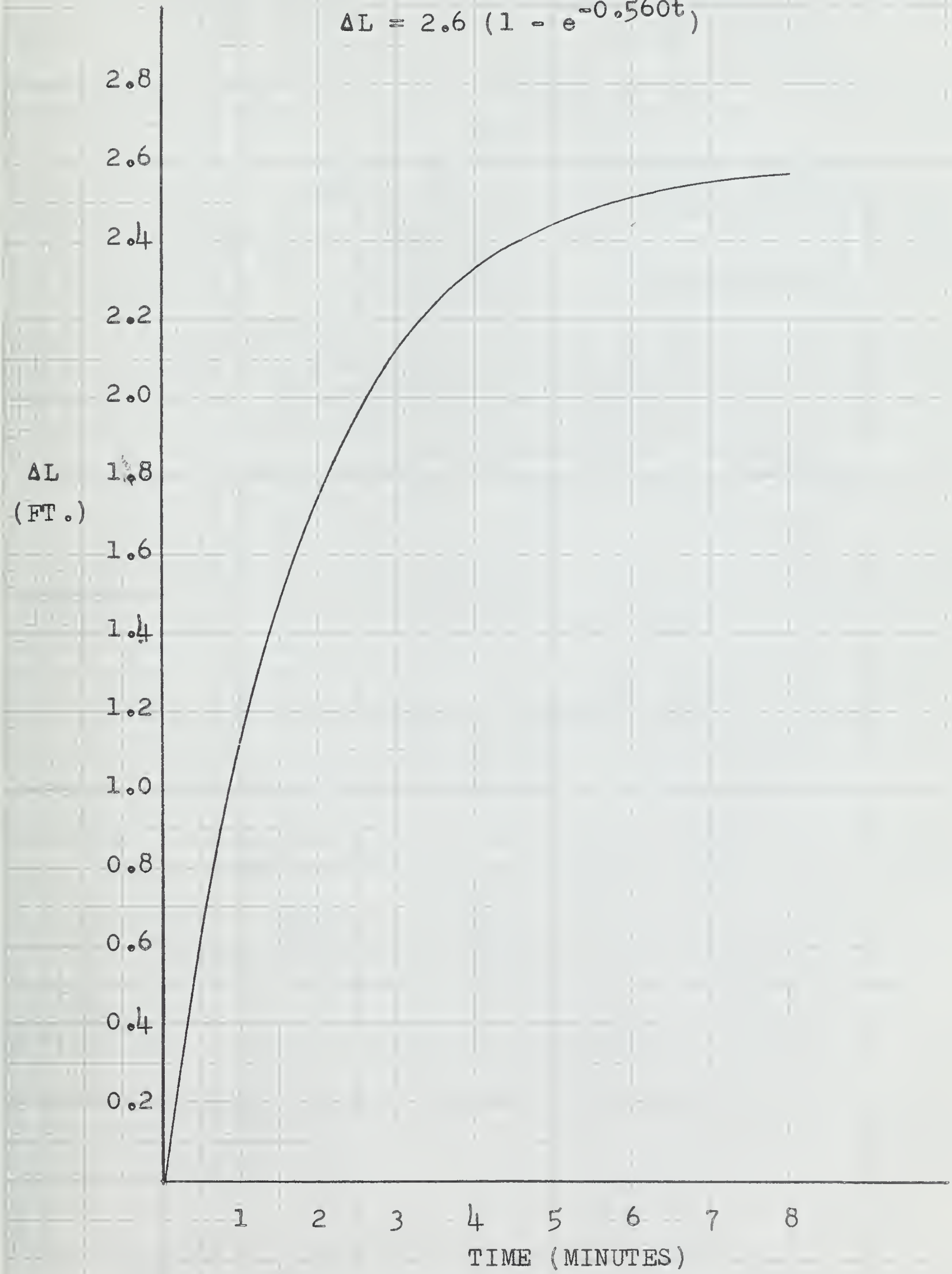


FIGURE 16

TEMPERATURE INCREASE FROM STEP

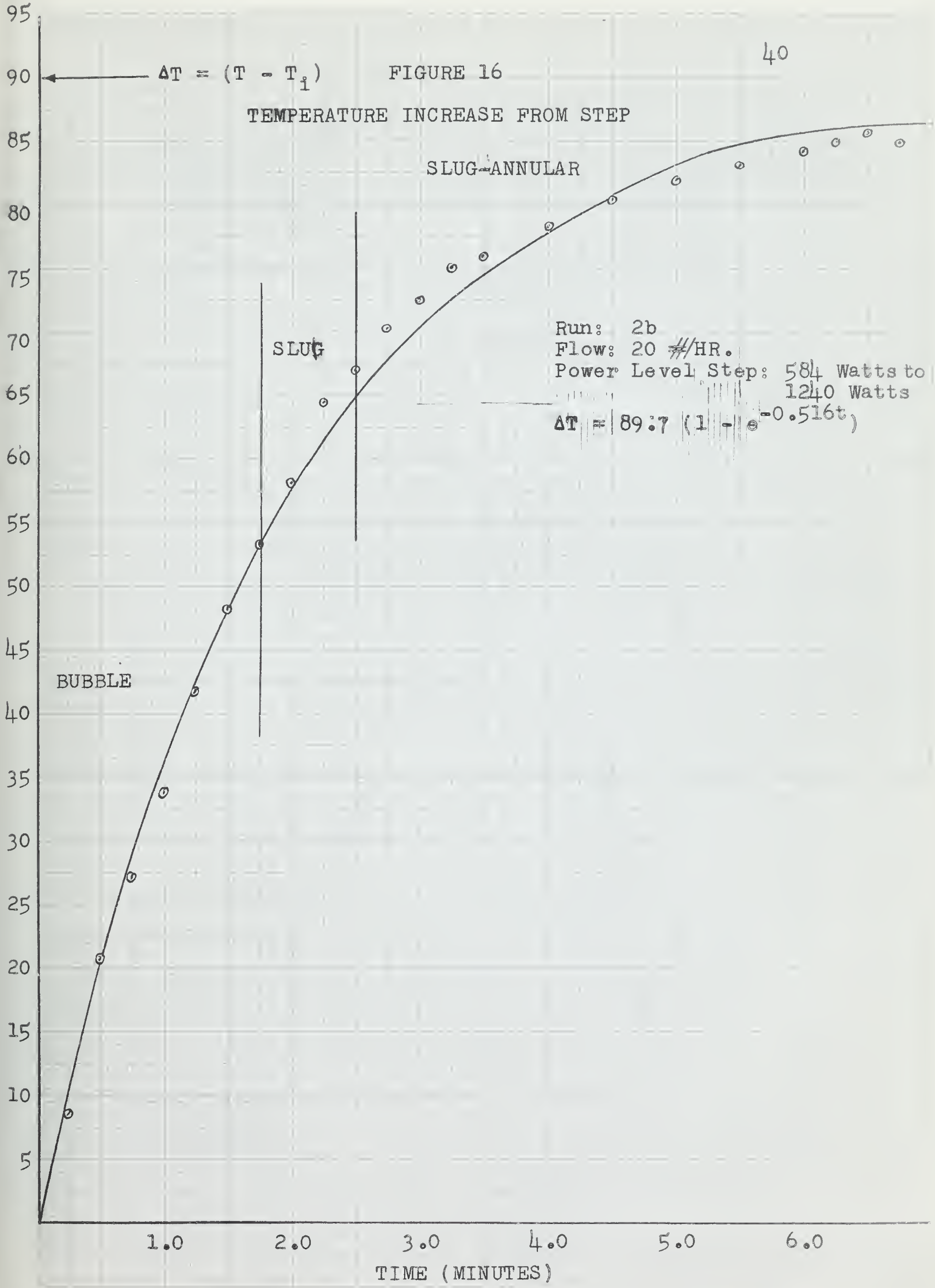


FIGURE 17

LOCATION PEAK WALL TEMPERATURE

Run: 2b
Flow: 20 #/HR.
Power Level Step: 584 Watts to 1240 Watts
ΔL_∞ = -2.50 FT.
Scale: 1" = 1'
__ __ __ = Conditions Before Step

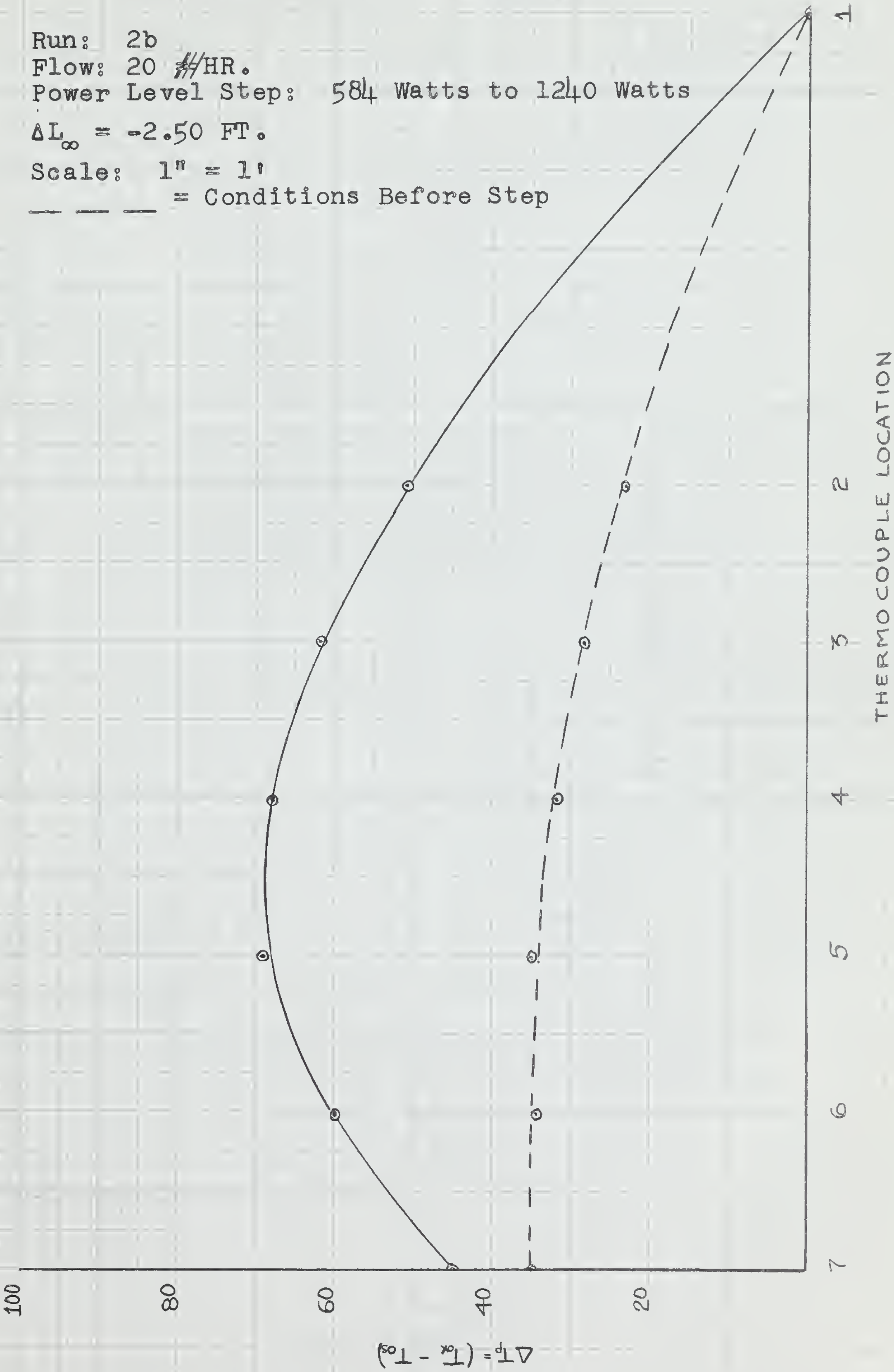
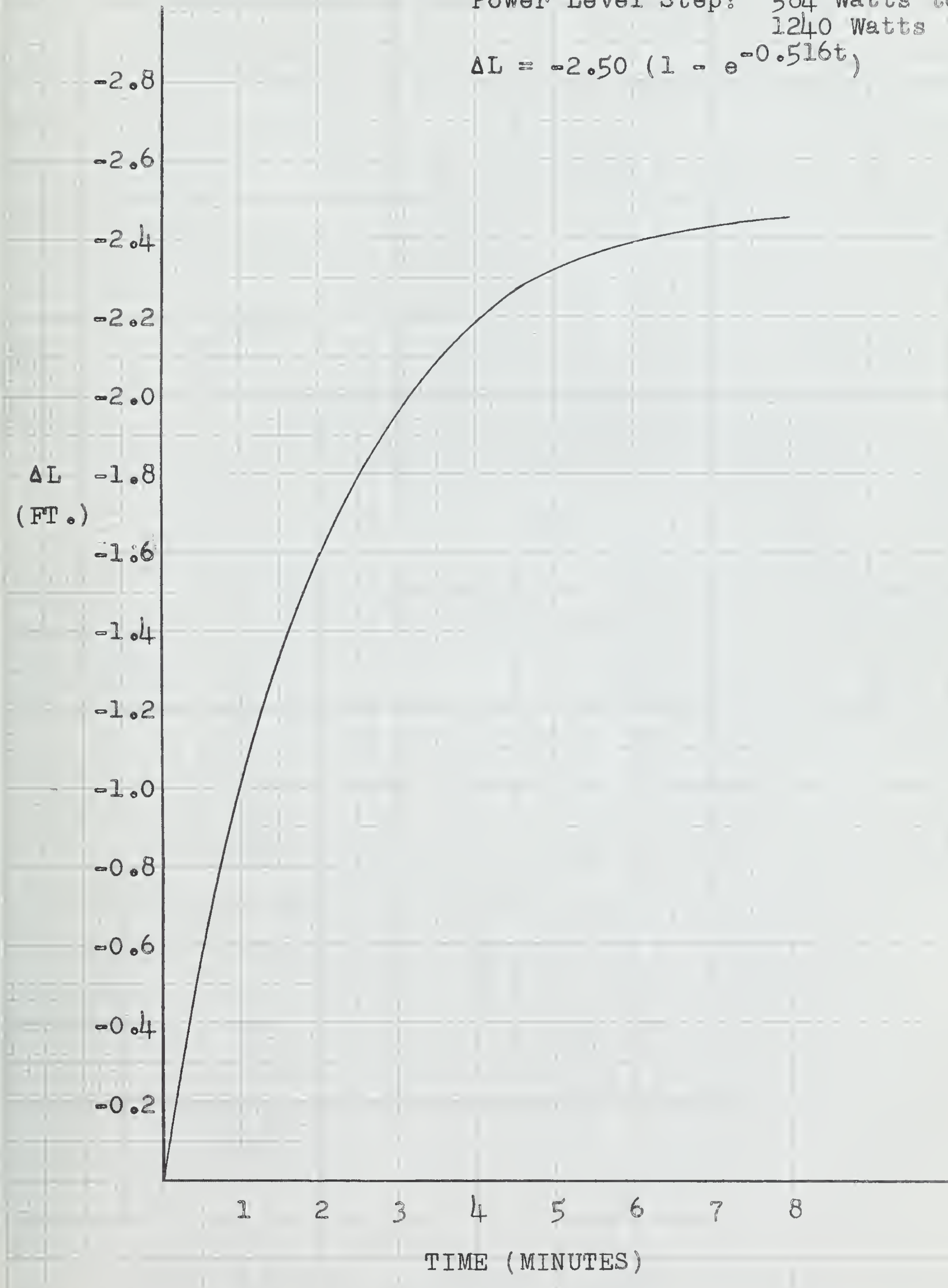


FIGURE 18

TOTAL DERIVED INTERFACE MOVEMENT

Run: 2b
Flow: 20 #/HR.
Power Level Step: 584 Watts to 1240 Watts

$\Delta L = -2.50\,(1 - e^{-0.516t})$



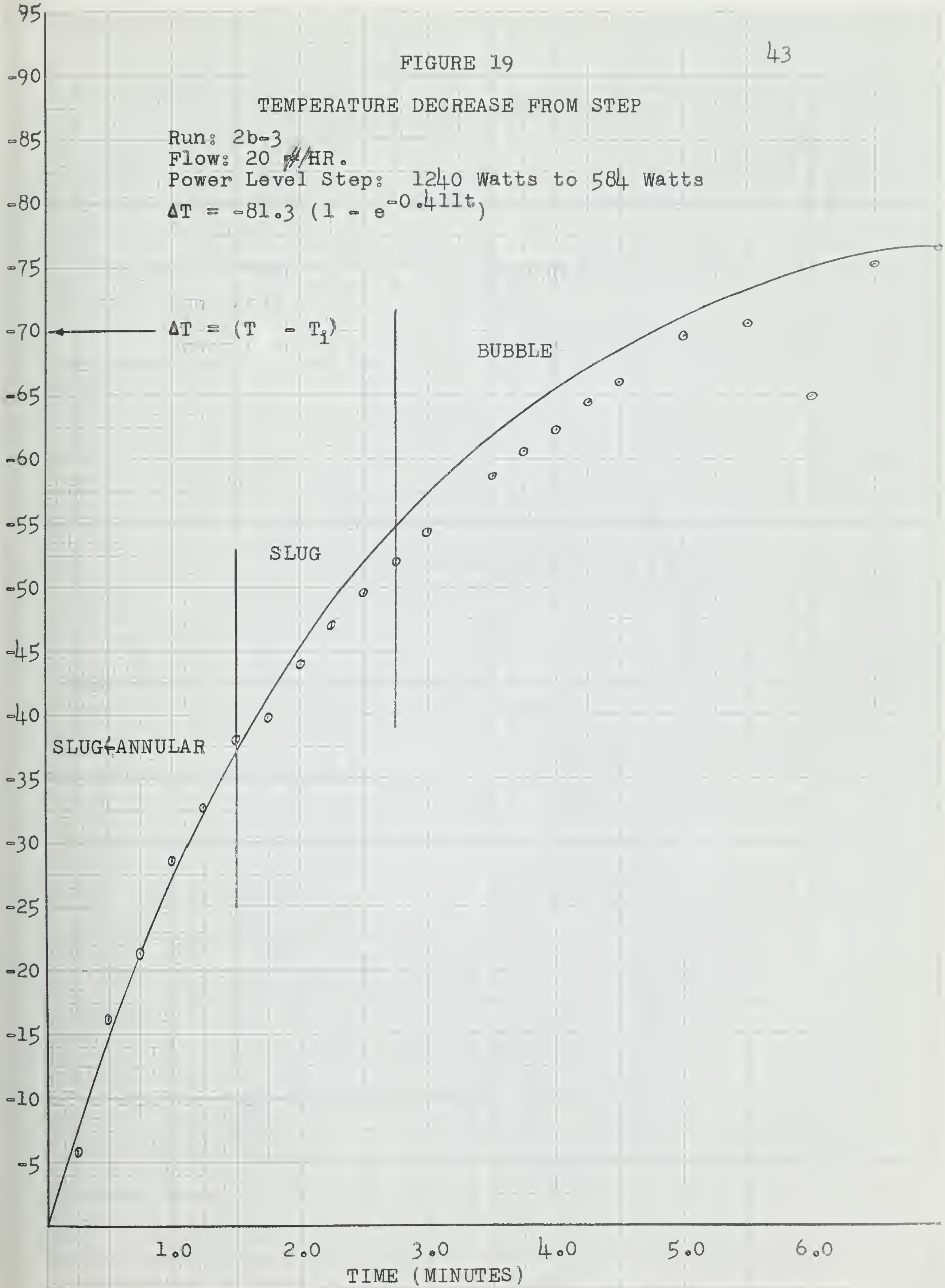


FIGURE 19
TEMPERATURE DECREASE FROM STEP
Run: 2b-3
Flow: 20 #/HR.
Power Level Step: 1240 Watts to 584 Watts
ΔT = -81.3 (1 - e^-0.411t)
ΔT = (T - T_i)
SLUG-ANNULAR
SLUG
BUBBLE
95
-90
-85
-80
-75
-70
-65
-60
-55
-50
-45
-40
-35
-30
-25
-20
-15
-10
-5
1.0
2.0
3.0
4.0
5.0
6.0
TIME (MINUTES)

FIGURE 20

LOCATION PEAK WALL TEMPERATURE

Run: 2b-3
Flow: 20 #/HR.
Power Level Step: 1240 Watts to 584 Watts
ΔL_{∞} = 2.50 FT.
Scale: 1" = 1'
— — — = Conditions Before Step

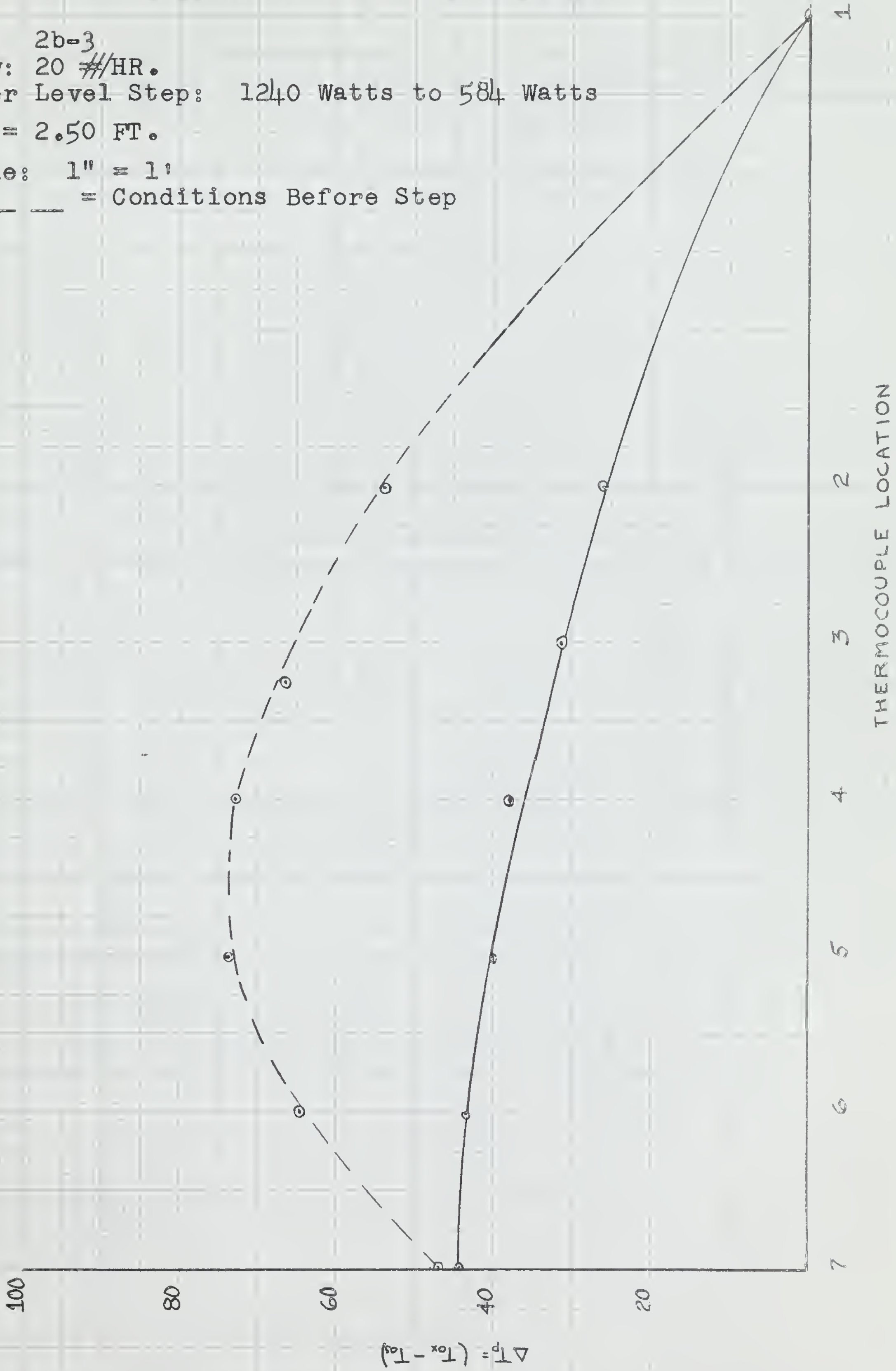
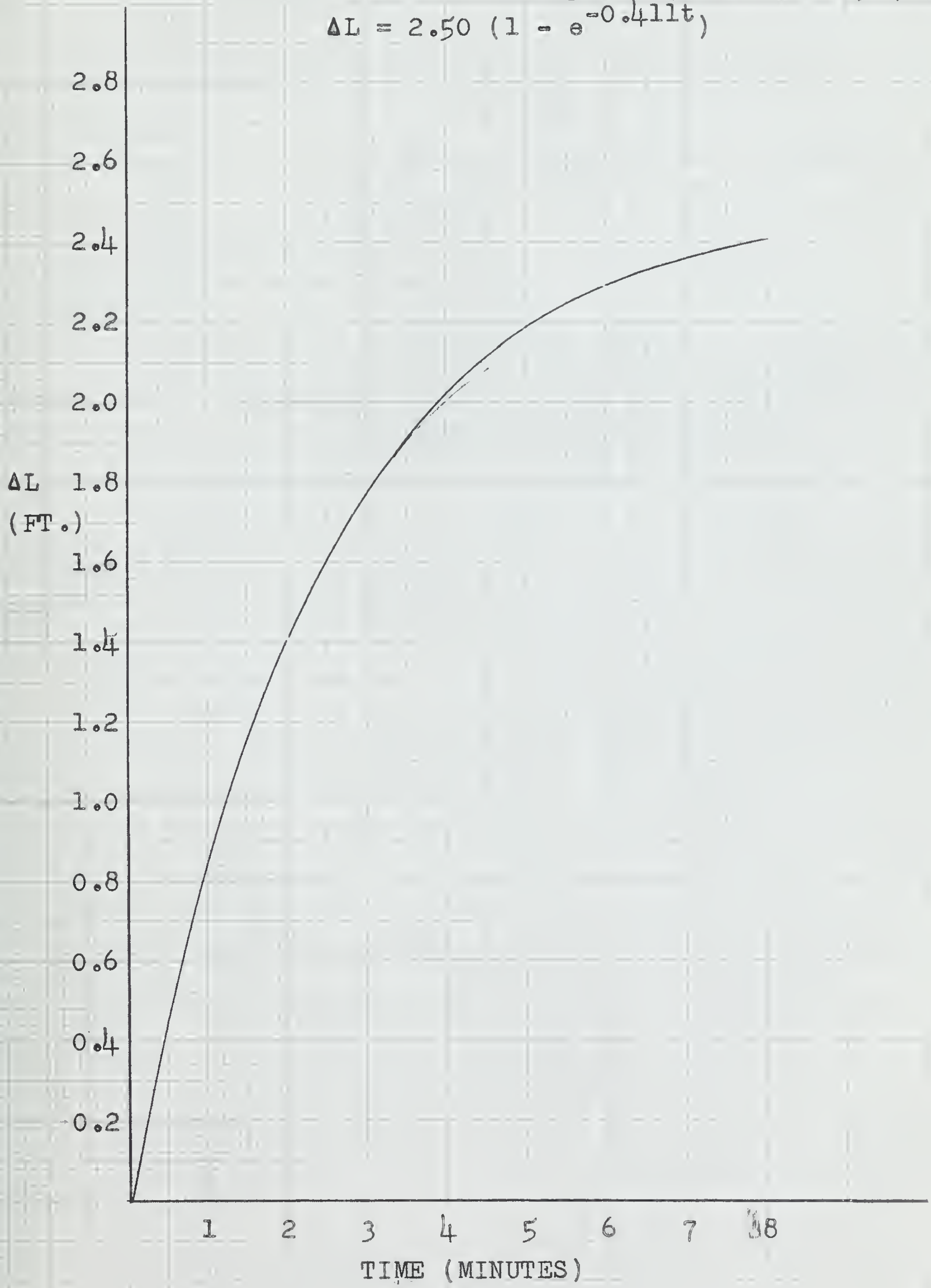


FIGURE 21

TOTAL DERIVED INTERFACE MOVEMENT

Run: 2b-3
Flow: 20 #/HR.
Power Level Step: 1240 Watts to 584 Watts
$\Delta L = 2.50\ (1 - e^{-0.411t})$



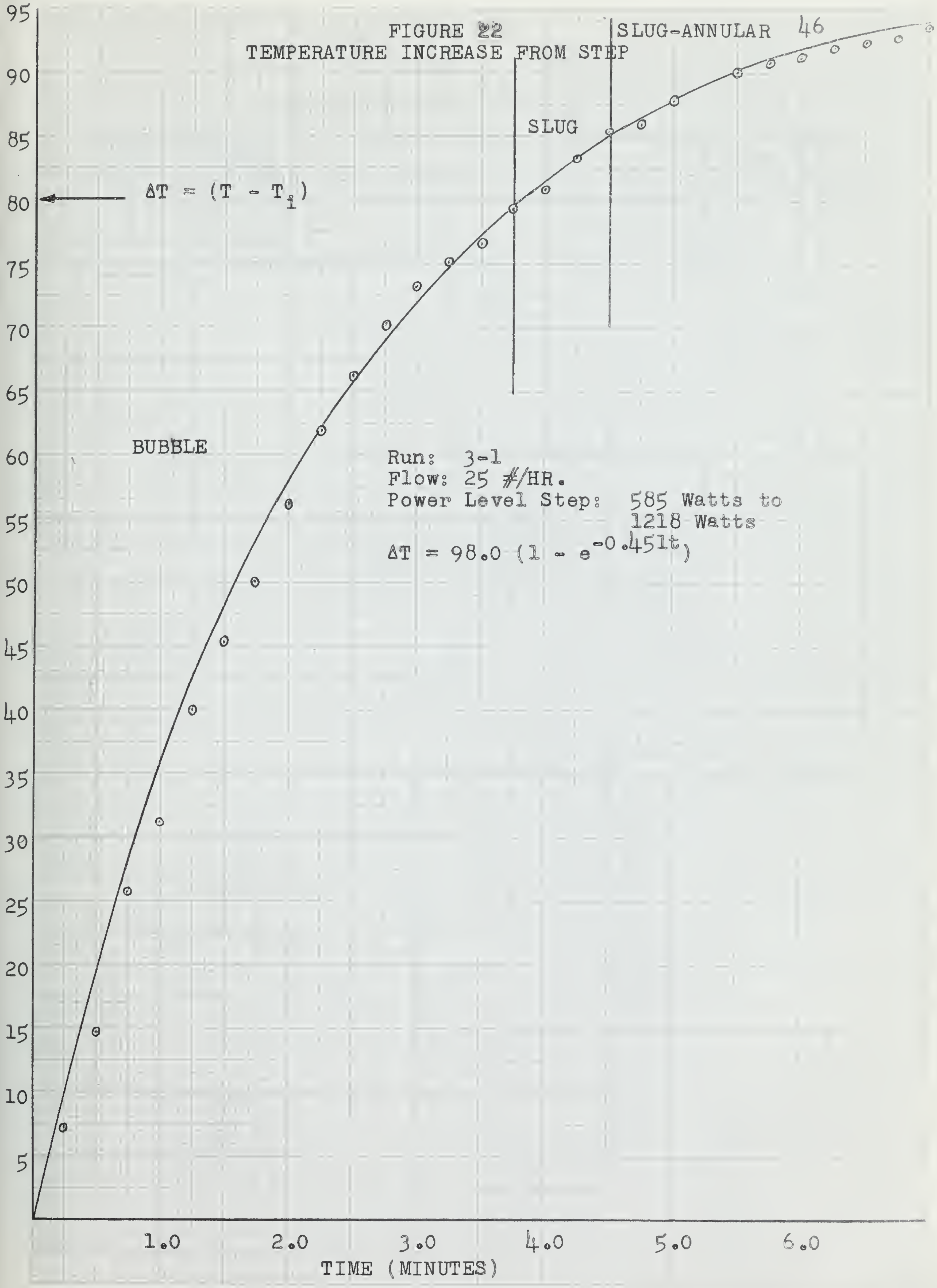


FIGURE 22
TEMPERATURE INCREASE FROM STEP
SLUG-ANNULAR
SLUG
BUBBLE
ΔT = (T - T_i)
Run: 3-1
Flow: 25 #/HR.
Power Level Step: 585 Watts to 1218 Watts
ΔT = 98.0 (1 - e^-0.451t)
TIME (MINUTES)
95
90
85
80
75
70
65
60
55
50
45
40
35
30
25
20
15
10
5
1.0
2.0
3.0
4.0
5.0
6.0

FIGURE 23

LOCATION PEAK WALL TEMPERATURE

Run: 3-1
Flow: 25 #/HR.
Power Level Step: 585 Watts to 1218 Watts
ΔL_{∞} = -2.30 FT.
Scale: 1" = 1'
— — — = Conditions Before Step

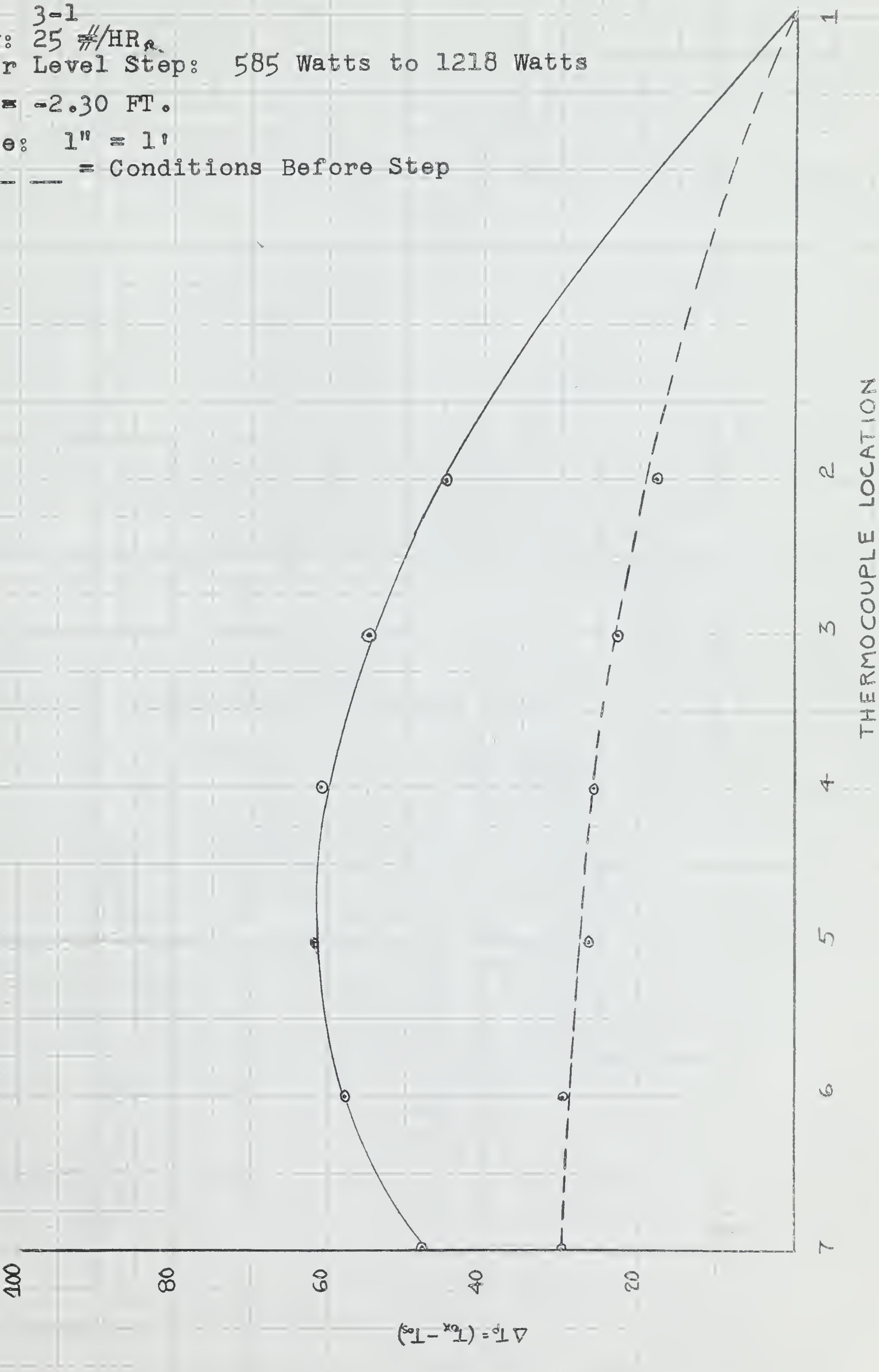
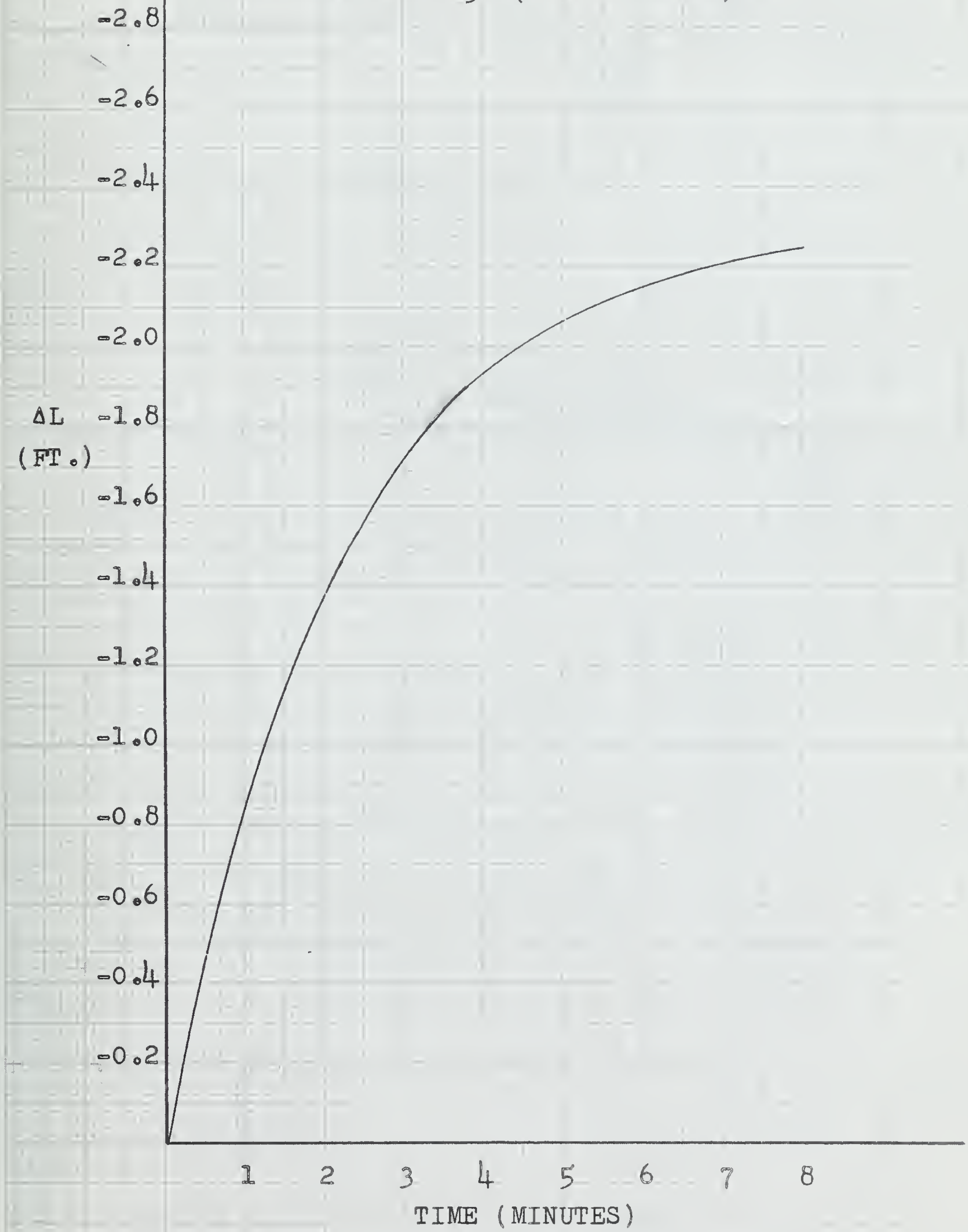


FIGURE 24

TOTAL DERIVED INTERFACE MOVEMENT

Run: 3-1
Flow: 25 #/HR.
Power Level Step: 585 Watts to 1218 Watts
$\Delta L = -2.30\ (1 - e^{-0.451t})$



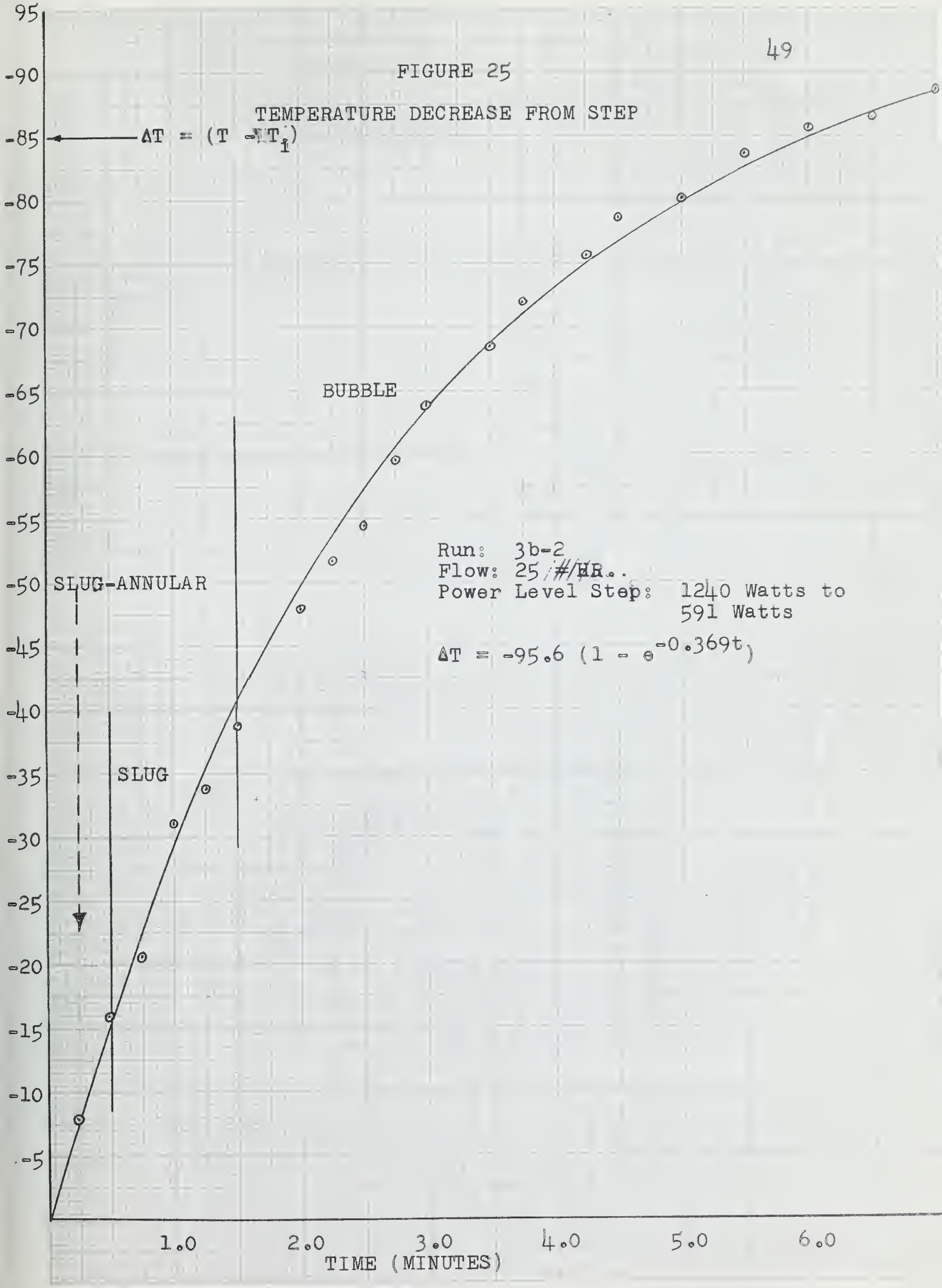


FIGURE 25
TEMPERATURE DECREASE FROM STEP
ΔT = (T - T_i)
95
-90
-85
-80
-75
-70
-65
-60
-55
-50
-45
-40
-35
-30
-25
-20
-15
-10
-5
BUBBLE
SLUG-ANNULAR
SLUG
Run: 3b-2
Flow: 25 #/HR.
Power Level Step: 1240 Watts to
591 Watts
ΔT = -95.6 (1 - e^{-0.369t})
1.0
2.0
3.0
4.0
5.0
6.0
TIME (MINUTES)

FIGURE 26

LOCATION PEAK WALL TEMPERATURE

Run: 3b-2
Flow: 25 #/HR.
Power Level Step: 1240 Watts to 591 Watts

ΔL_{∞} = 2.30 FT.

Scale: 1" = 1'

— — — = Conditions Before Step

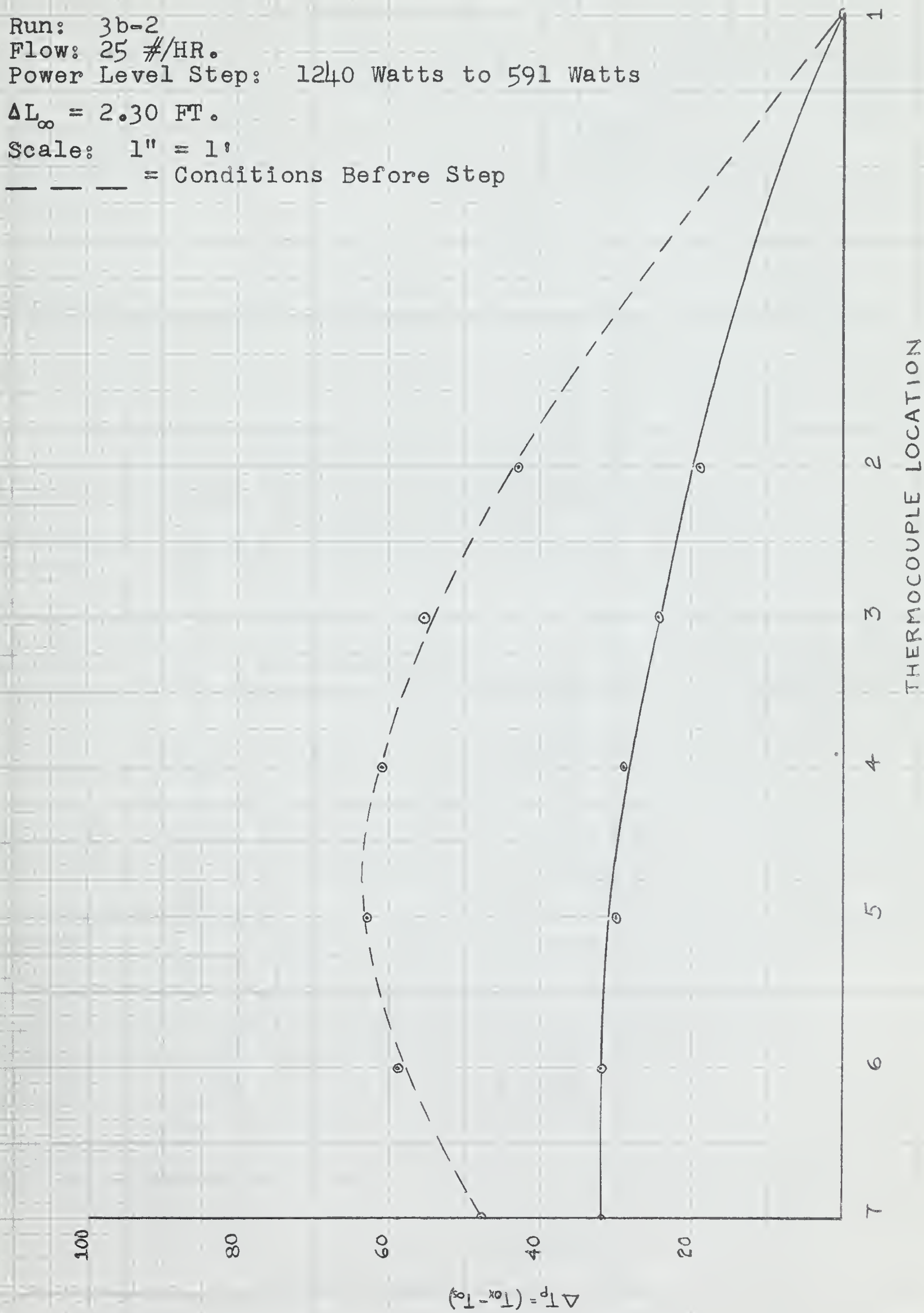


FIGURE 27

TOTAL DERIVED INTERFACE MOVEMENT

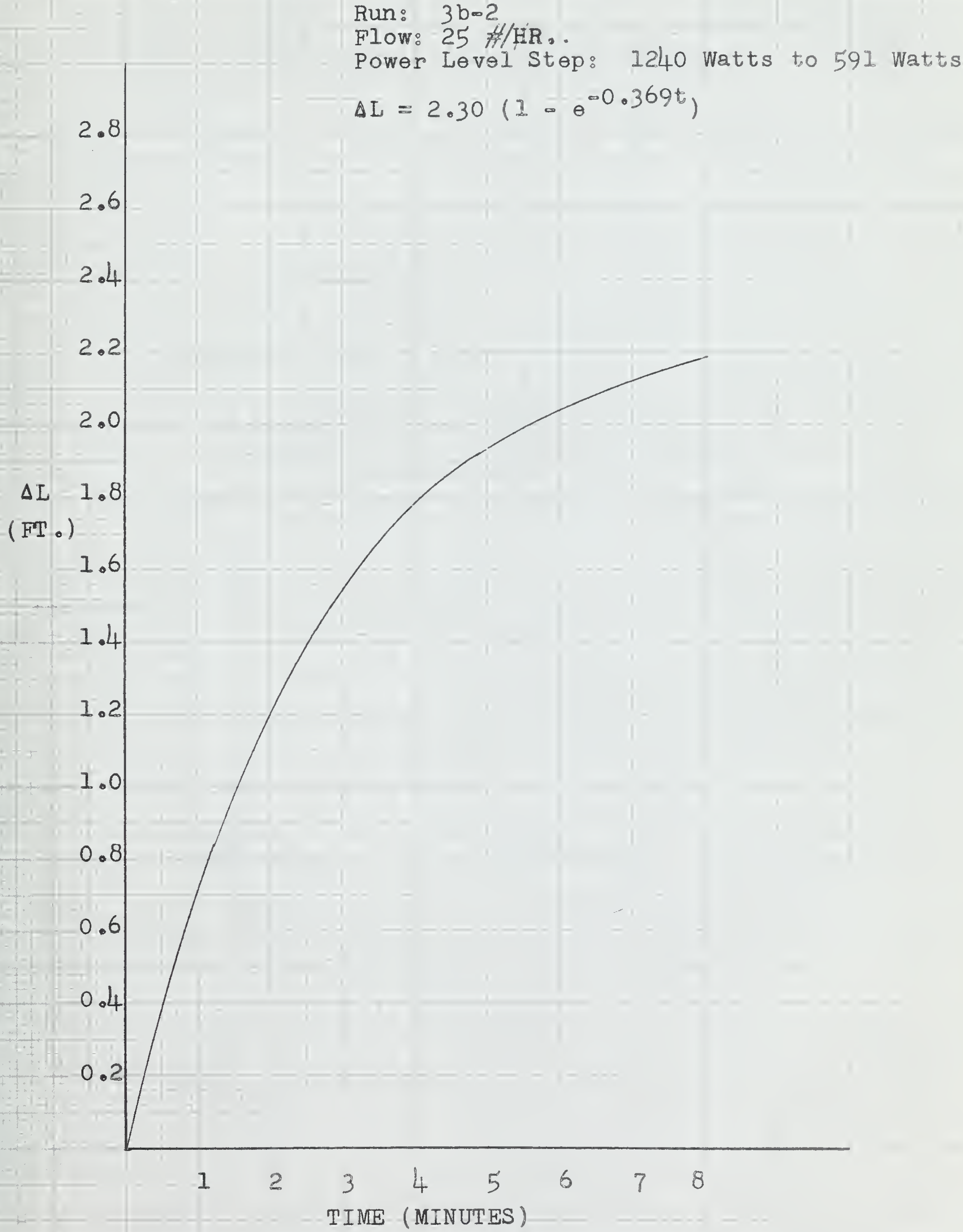


FIGURE 28

TEMPERATURE INCREASE FROM STEP

Run: 101-b
Power Level: 986 Watts
Flow Step: 25 #/HR. to 16.5 #/HR.

$\Delta T = 12.7\,(1 - e^{-0.435t})$

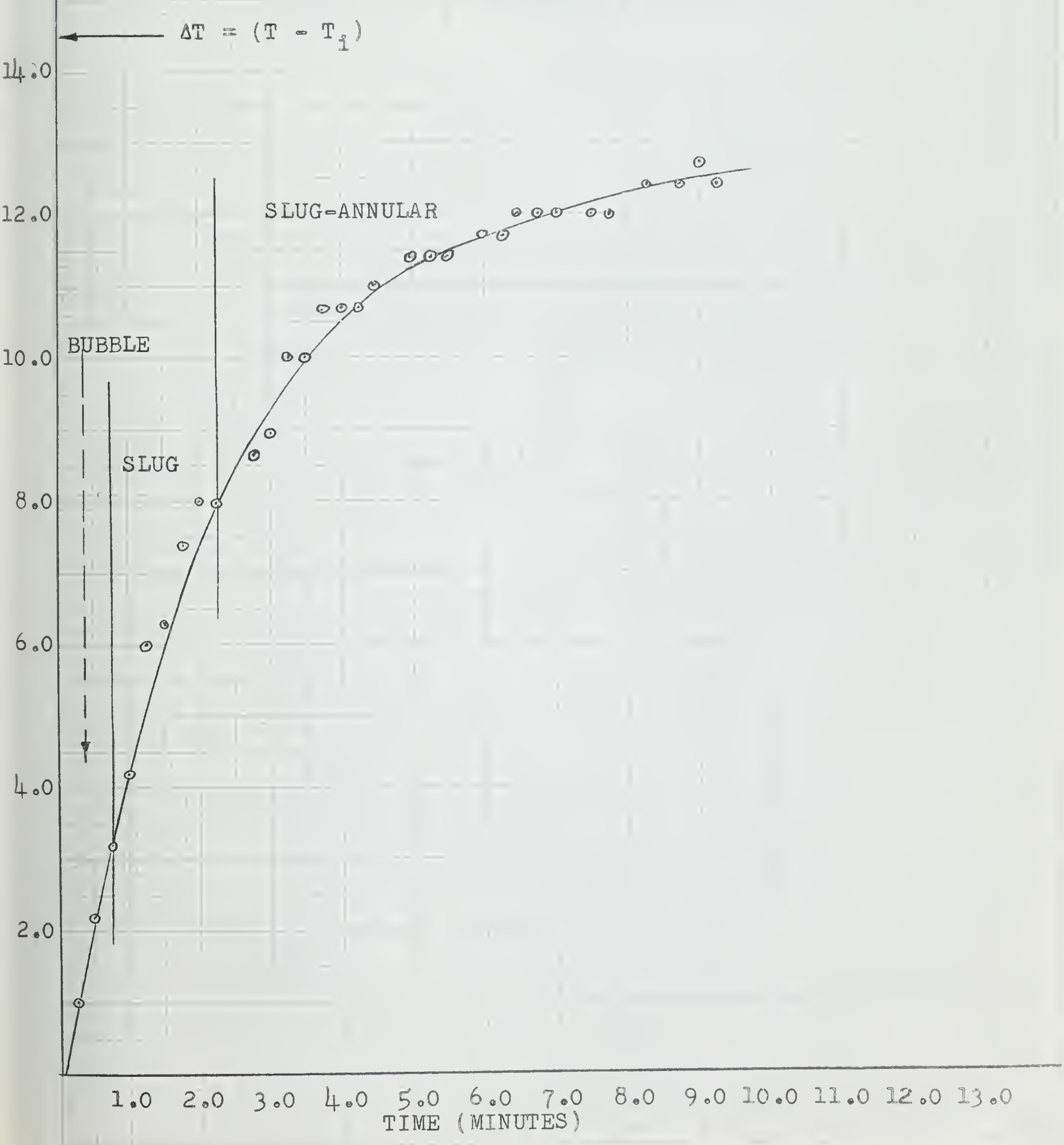


FIGURE 29

LOCATION PEAK WALL TEMPERATURE

Run: 101-b
Power Level: 986 Watts
Flow Step: 25 #/HR. to 16.5 #/HR.

ΔL_∞ = -2.20 FT.

Scale: 1" = 1'

— — — = Conditions Before Step

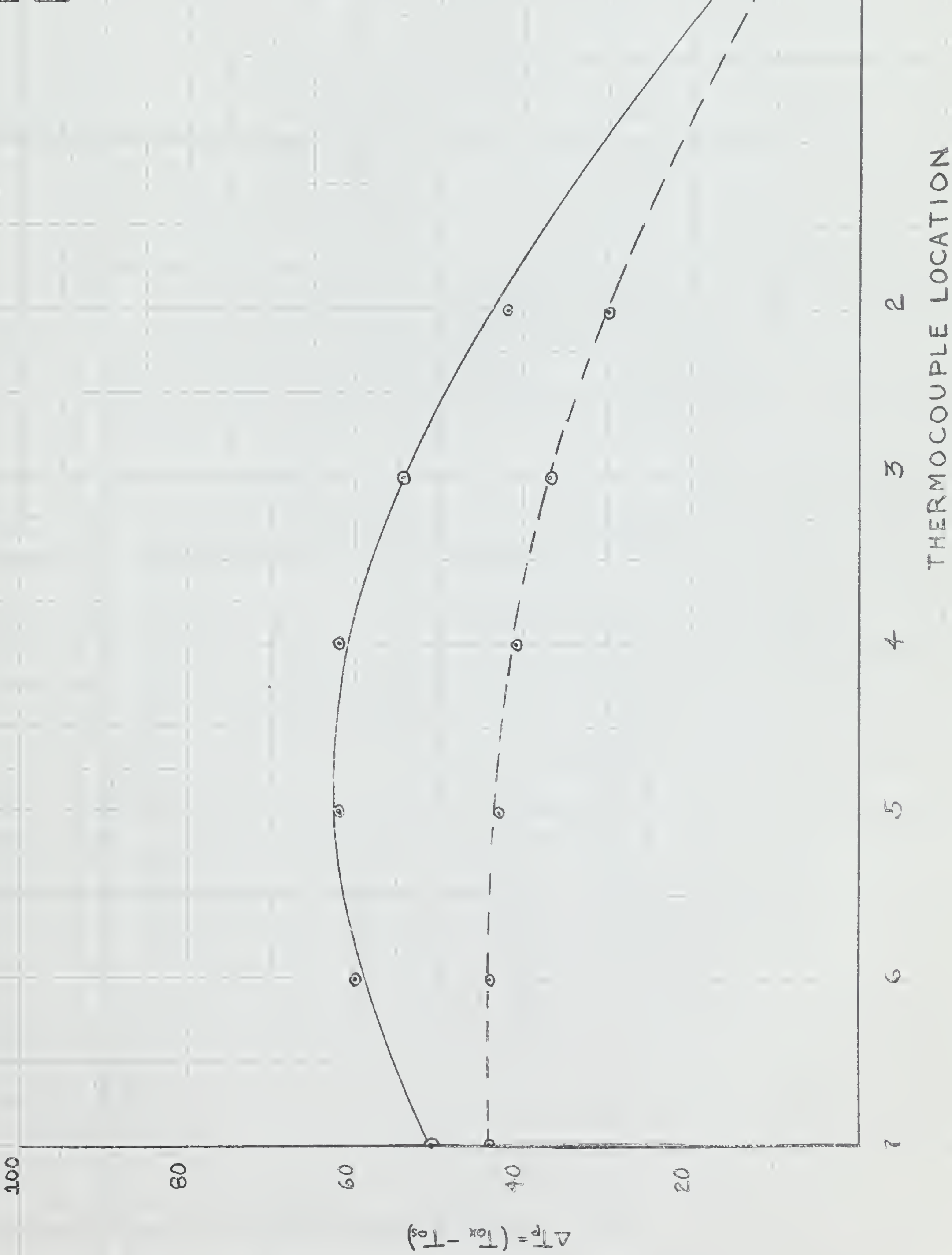


FIGURE 30

TOTAL DERIVED INTERFACE MOVEMENT

Run: 101-b
Power Level: 986 Watts
Flow Step: 25 #/HR. to 16.5 #/HR.

$\Delta L = -2.20\,(1 - e^{-0.435t})$

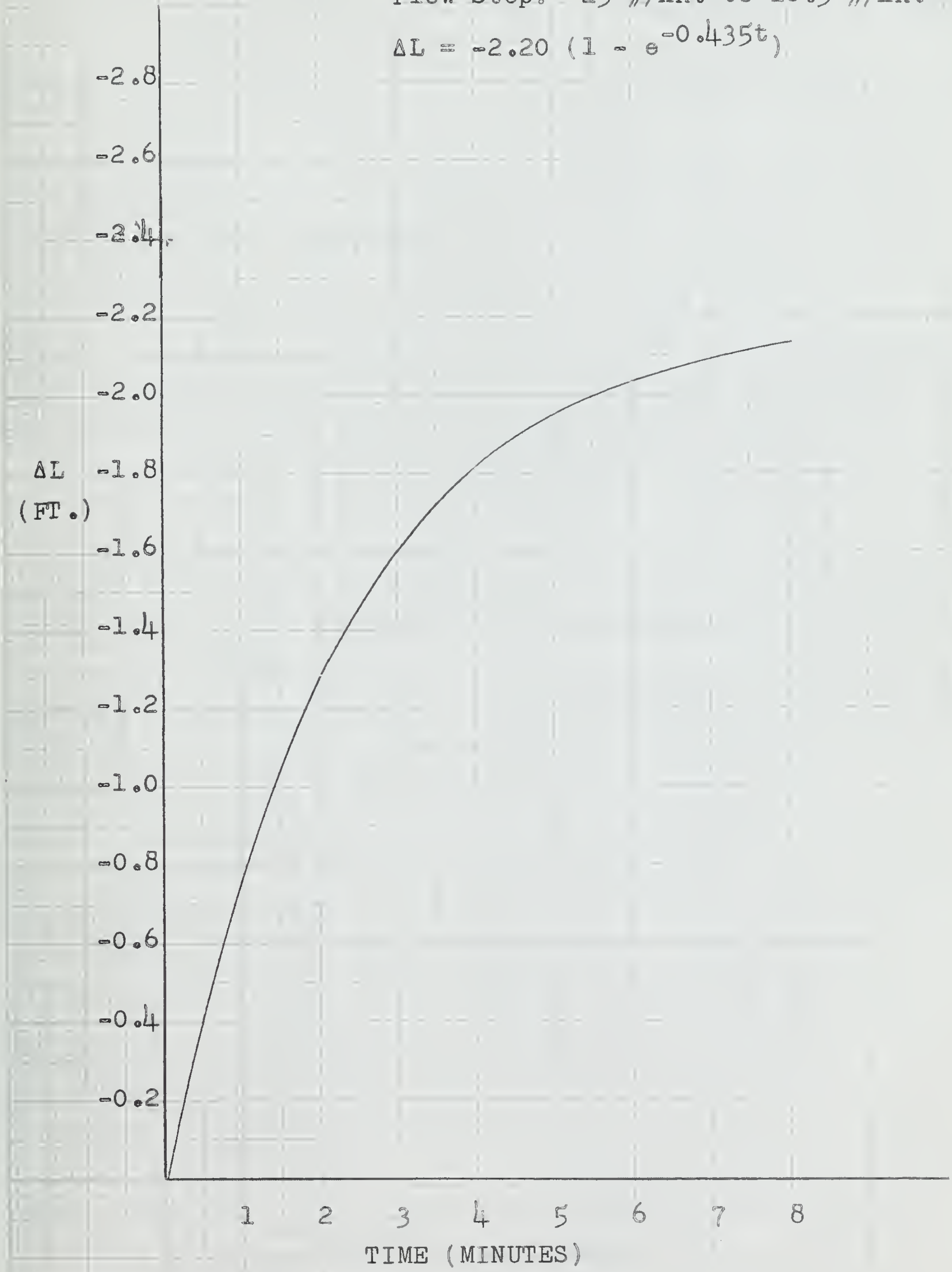


FIGURE 31

TEMPERATURE DECREASE FROM STEP

Run: 101-c
Power Level: 966 Watts
Flow Step: 16.5 #/HR. to 25.0 #/HR.

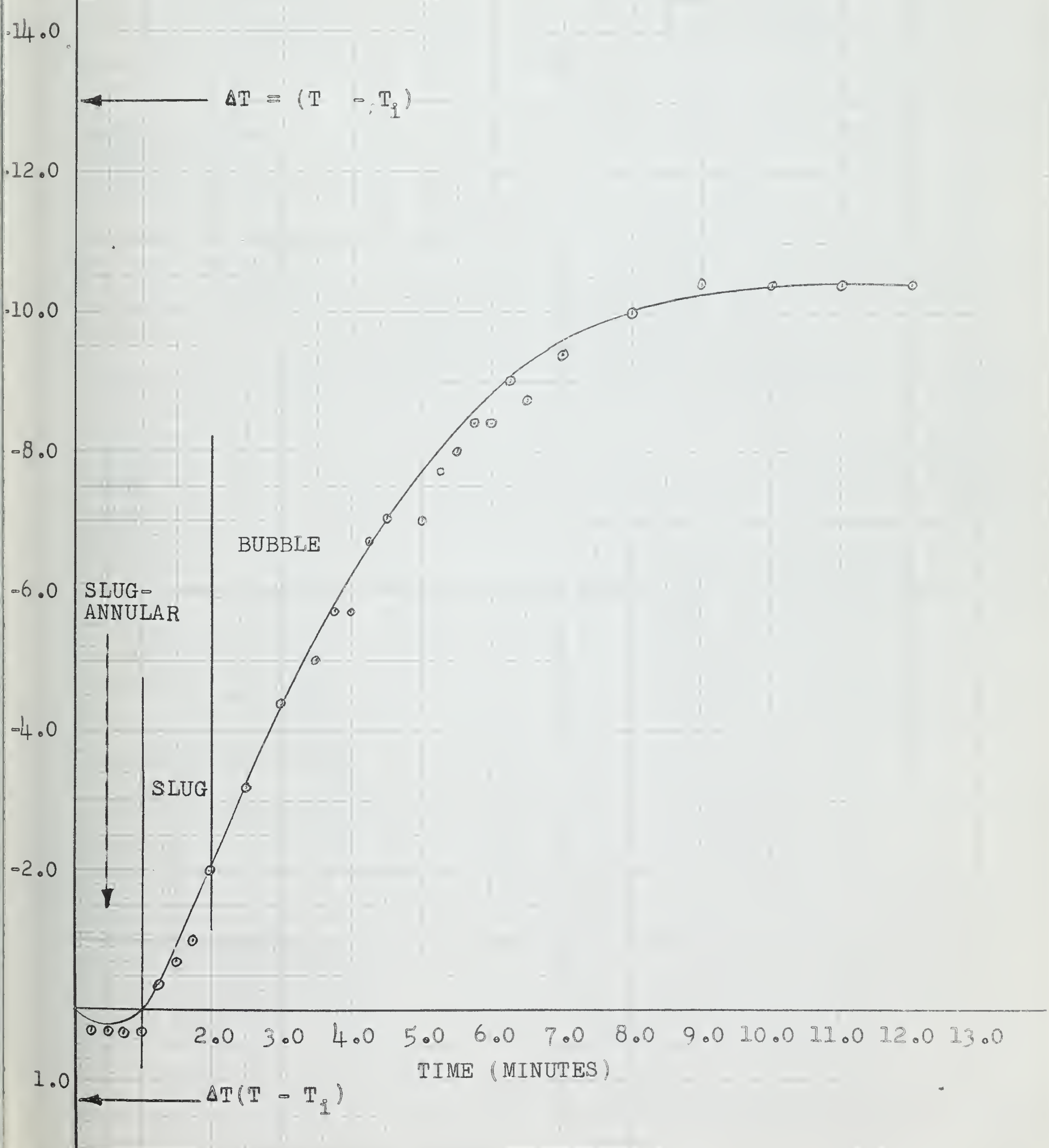


FIGURE 32

LOCATION PEAK WALL TEMPERATURE

Run: 101c
Power Level: 966 Watts
Flow Step: 16.5 #/HR. to 25.0 #/HR.

$\Delta L_{\infty} = 2.25$ FT.

Scale: 1" = 1'

— — — = Conditions Before Step

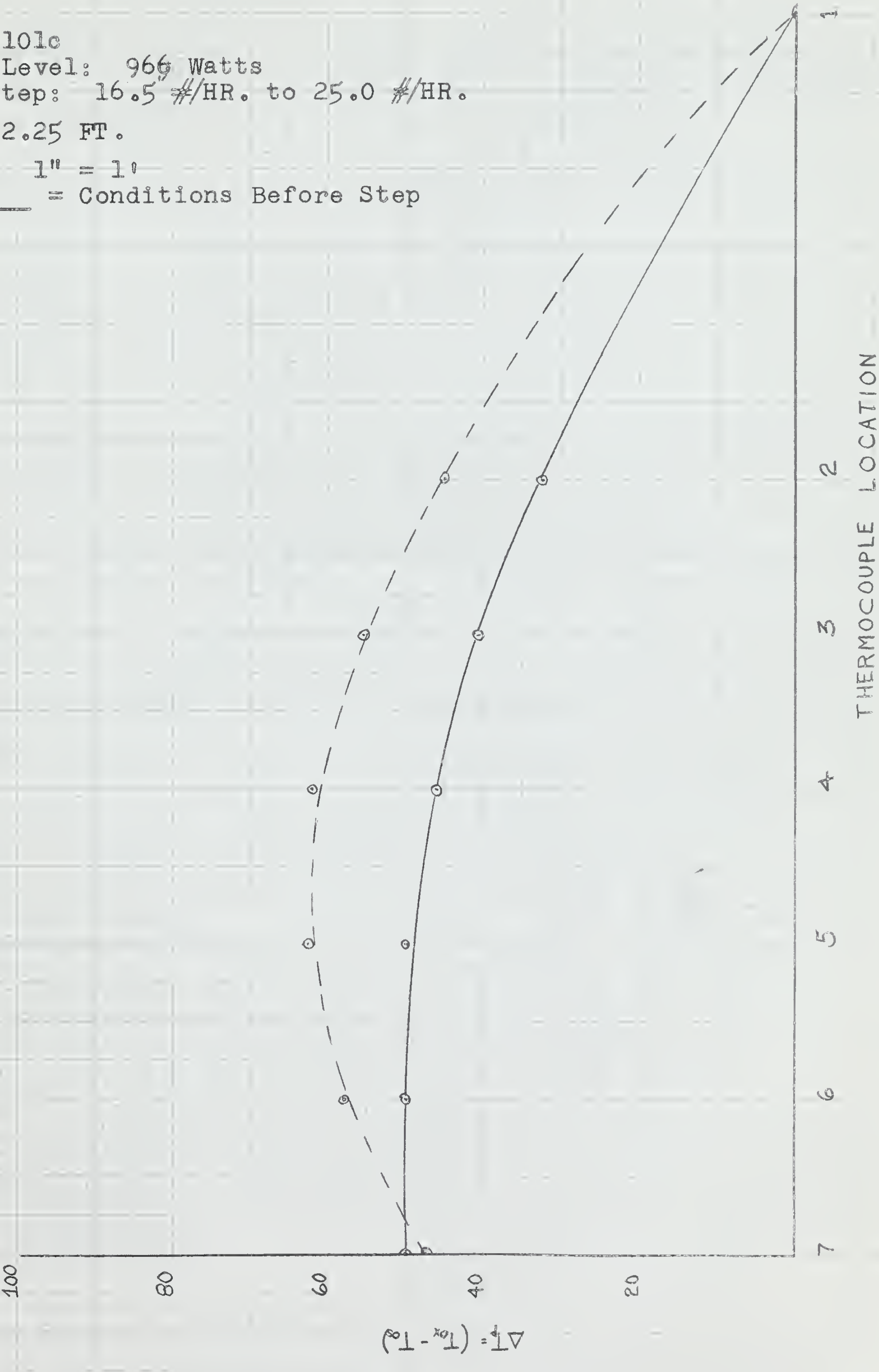


FIGURE 33

TEMPERATURE INCREASE FROM STEP

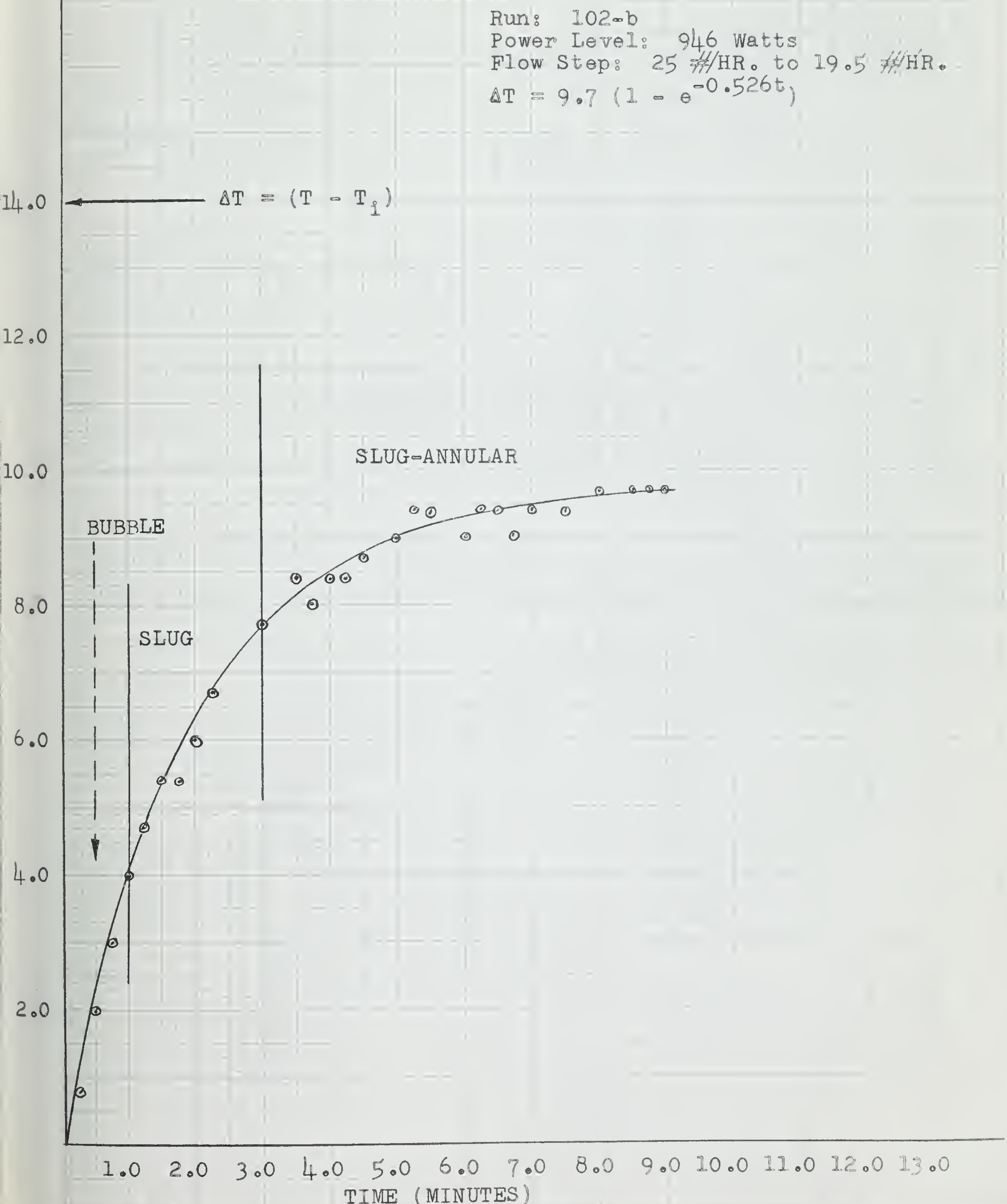


FIGURE 34

LOCATION PEAK WALL TEMPERATURE

Run: 102-b
Power Level: 946 Watts
Flow Step: 25 #/HR. to 19.5 #/HR.

$\Delta L_{\infty} = -2.15$ FT.

Scale: 1" = 1'

— — — = Conditions Before Step

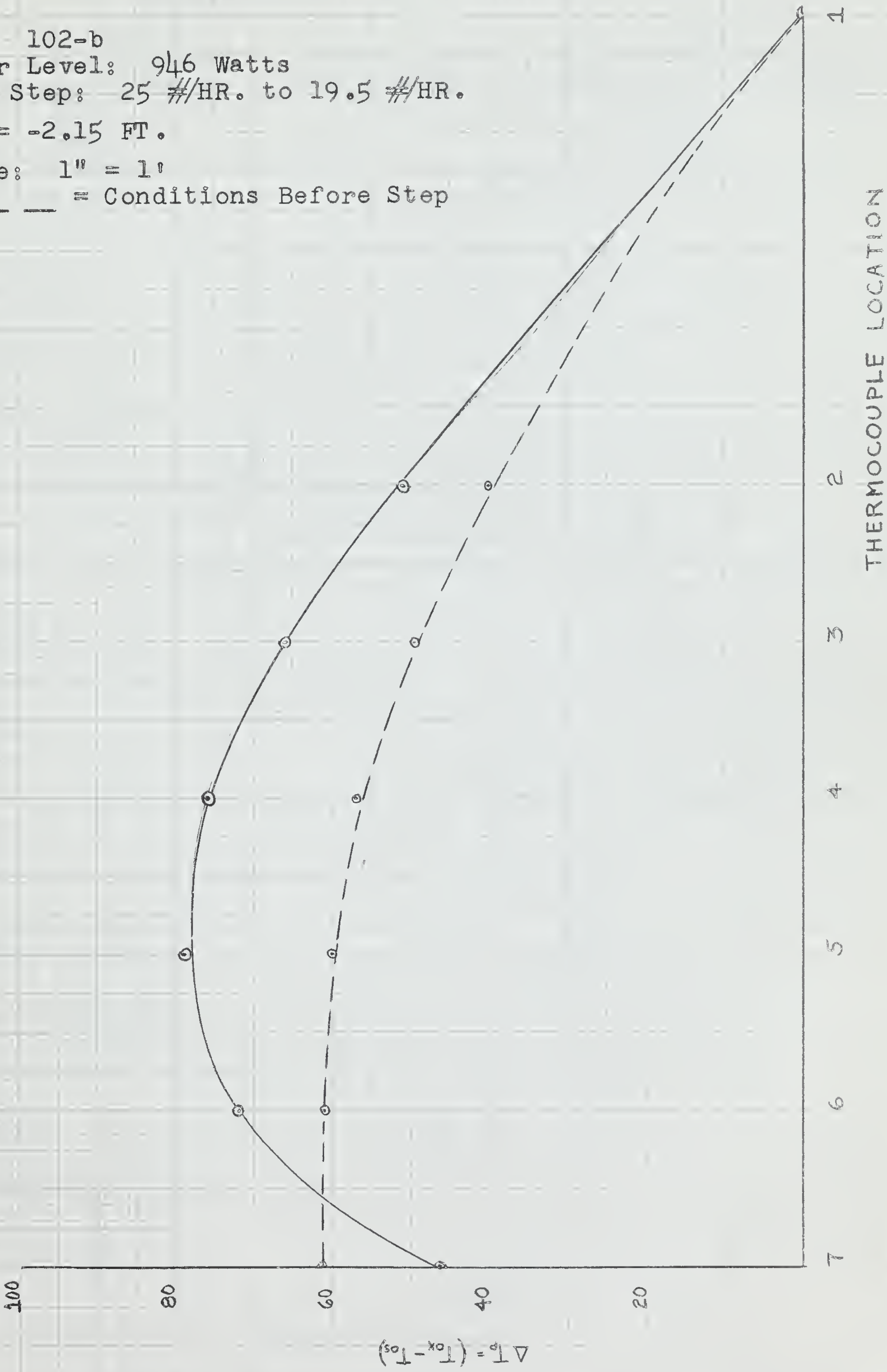


FIGURE 35

TOTAL DERIVED INTERFACE MOVEMENT

Run: 102-b
Flow Step: 25 #/HR. to 19.5 #/HR.

$\Delta L = -2.15\,(1 - e^{-0.526t})$

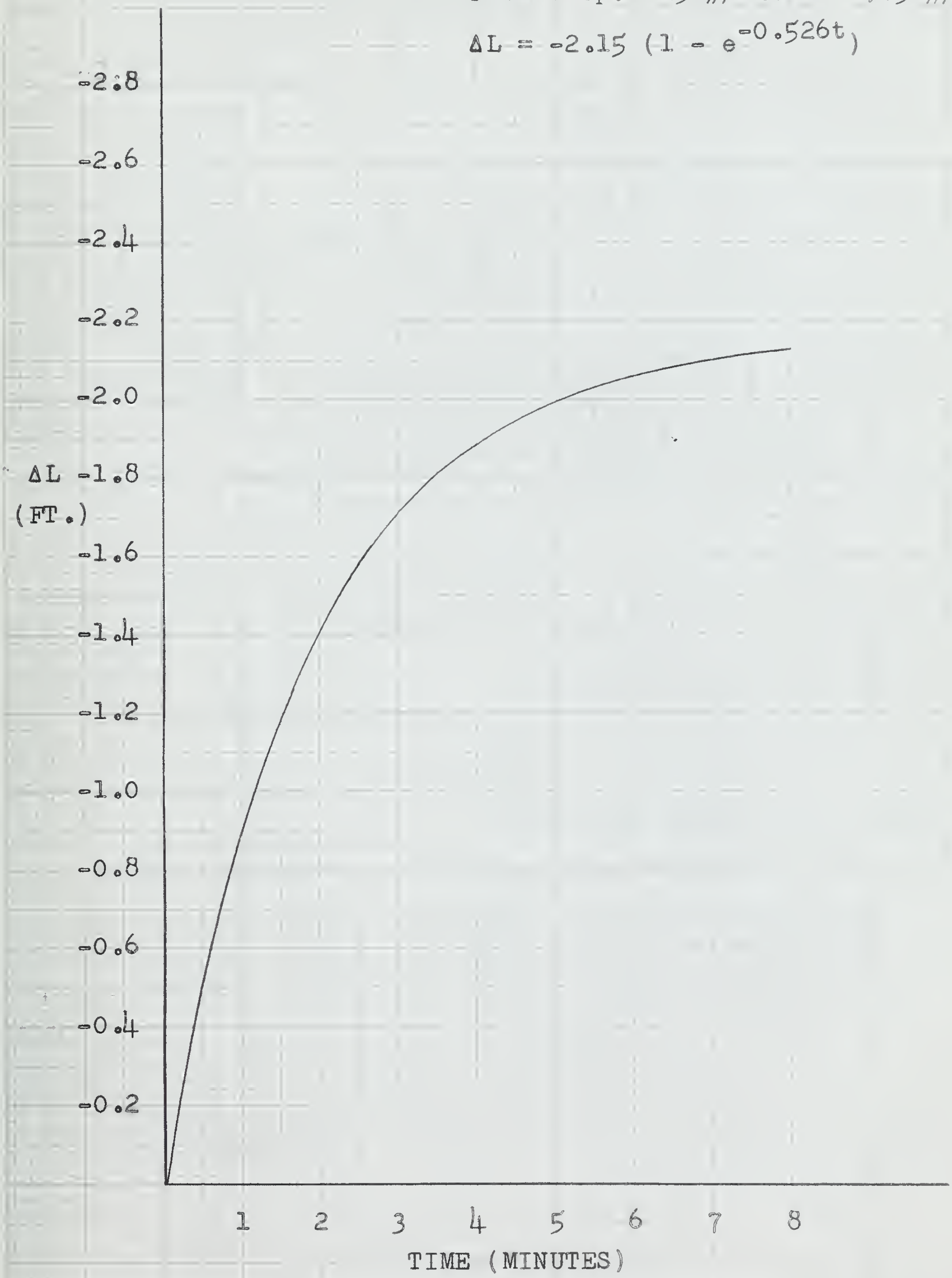


FIGURE 36

TRANSIENT SATURATED INTERFACE
VARIATION IN RESPONSE TO STEP
CHANGES IN HEAT FLOW RATE, q,
WITH WEIGHT FLOW RATE, W,
CONSTANT

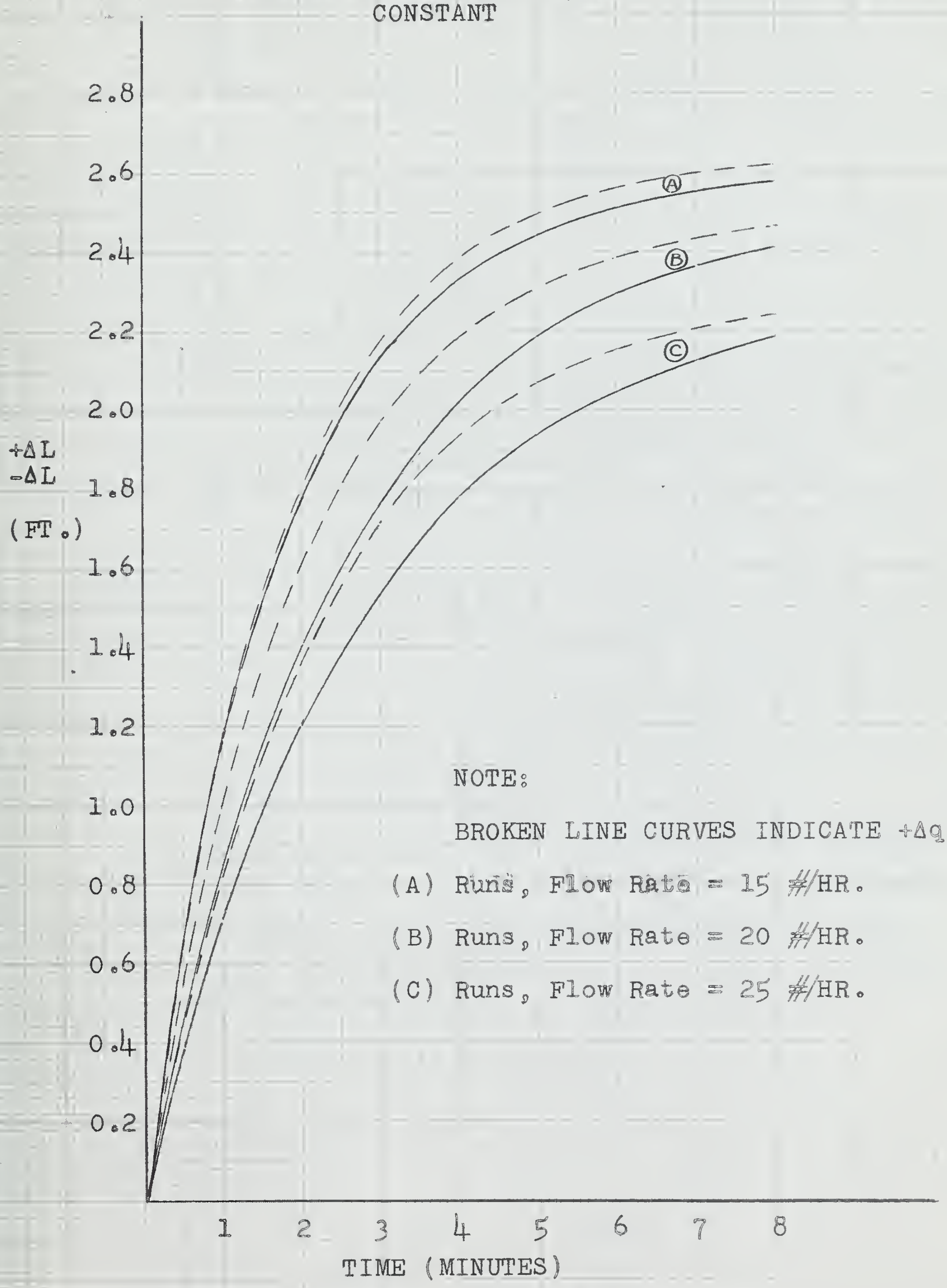


FIGURE 37

TRANSIENT SATURATED INTERFACE VARIATION IN RESPONSE TO STEP CHANGES IN WEIGHT FLOW RATE, W, WITH HEAT FLOW RATE, q, CONSTANT

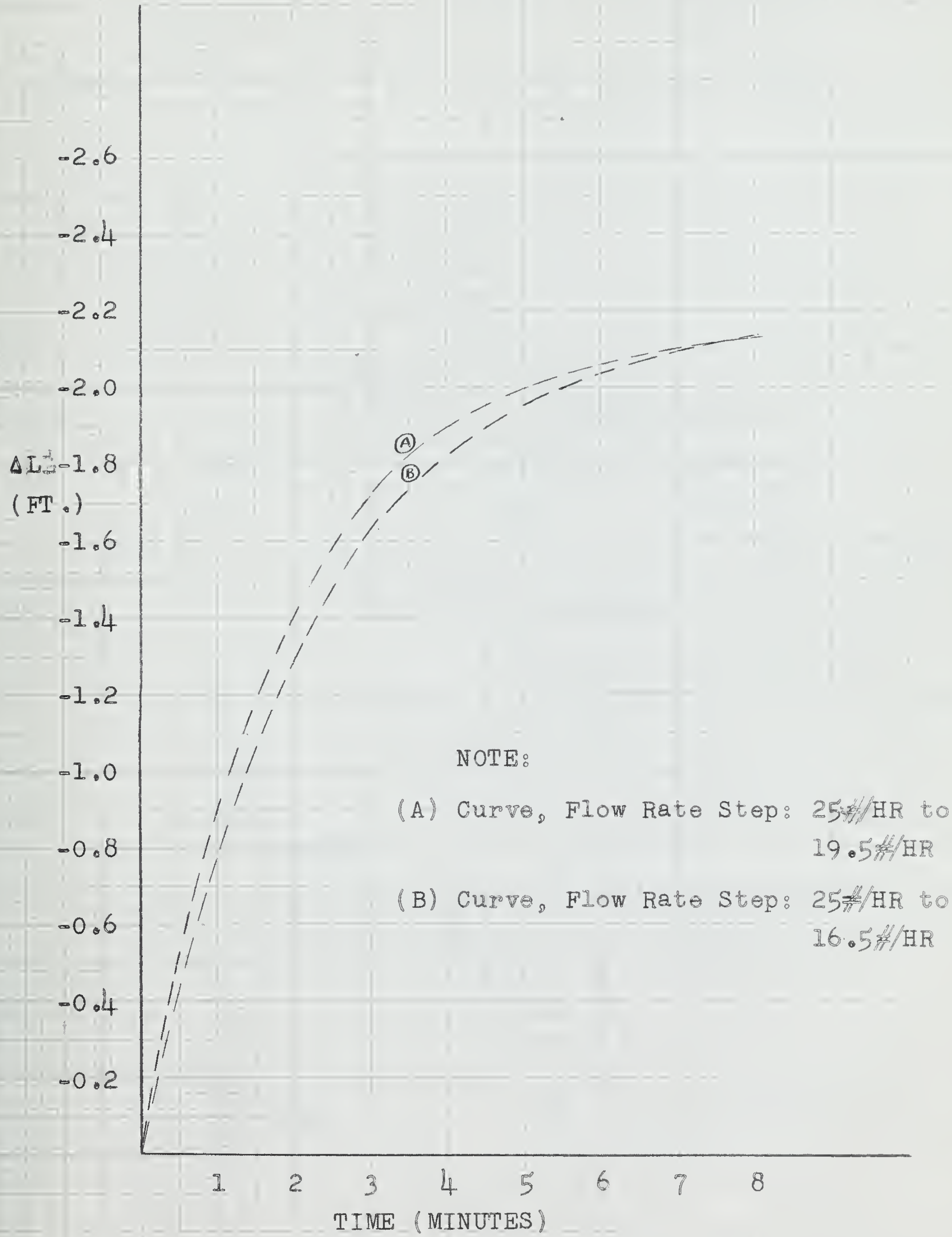


FIGURE 38

VARIATION OF "a" WITH WEIGHT FLOW RATE, W.

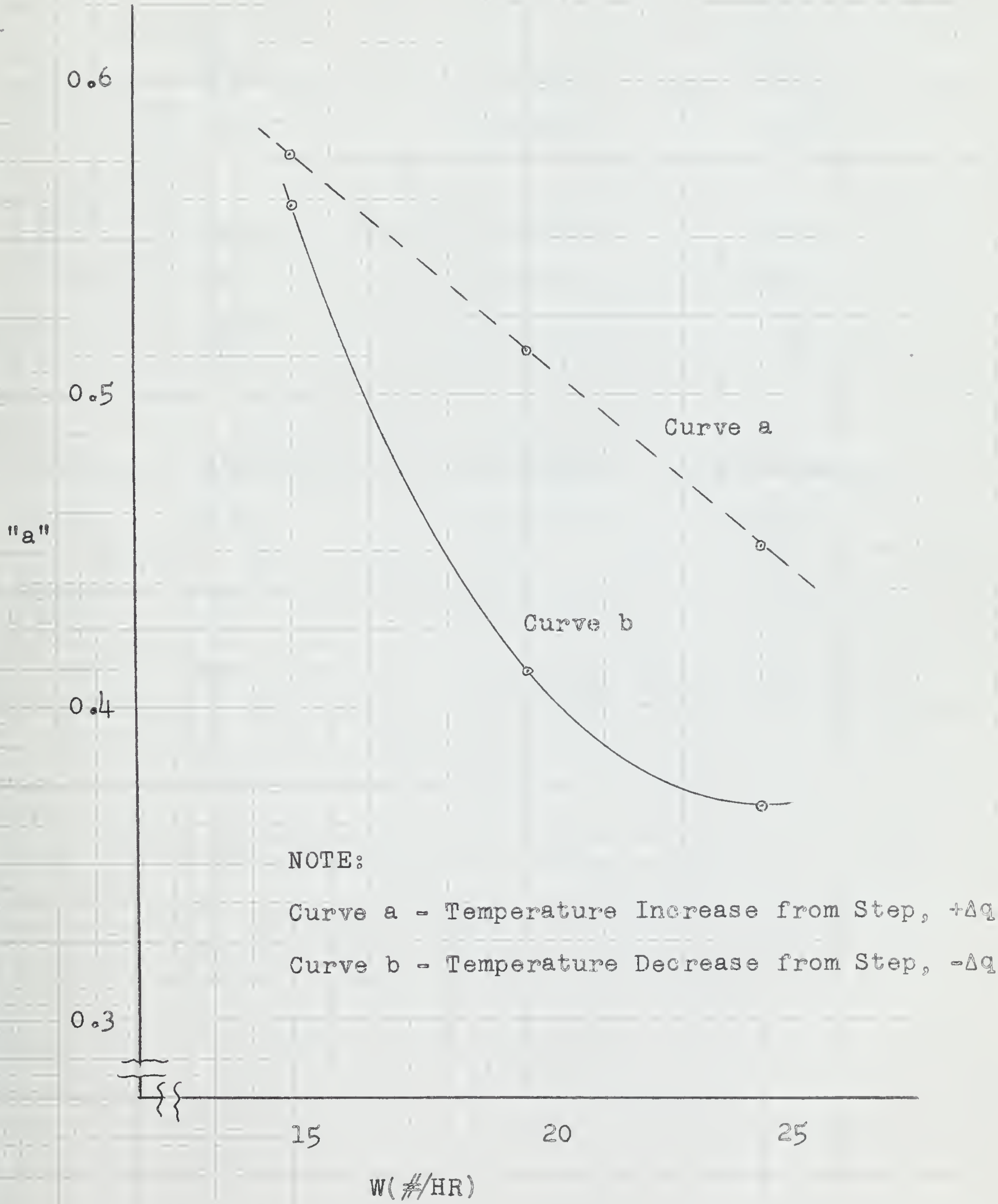


TABLE I

SUMMARY OF EXPERIMENTALLY DETERMINED ΔL_∞

RUN	W(#/HR)	Δq(WATTS)	ΔL_∞(FT)
1b-1	15.0	591-1240	2.65
2b	20.0	584-1240	2.50
3-1	25.0	585-1218	2.30
1b-1	15.0	1240-591	2.60
2b-3	20.0	1240-584	2.50
3b-2	25.0	1240-591	2.30

RUN	q(WATTS)	ΔW(#/HR)	ΔL_∞(FT)
101b	986	25.0-16.5	2.20
102b	946	25.0-19.5	2.15
101c	966	16.5-25.0	2.25

CHAPTER IV

DISCUSSION OF RESULTS

The results of the tests show that in all runs except those of Type 3 the saturated interface movement under transient conditions follows the general form of equation:

$$\Delta L = \Delta L_{\infty}(1 - e^{-at}) \tag{5}$$

The time constant of this equation, "a", is an indication of the time characteristics of the evaporator. In essence, it determines the initial slope of the transient response. FIGURES 36 and 37 indicate that "a" is strongly a function of the weight flow rate, W, heat flow rate, q, and the change in these two variables. FIG. 38 illustrates this marked variation. For a positive step change in heat flow rate, $+\Delta q$, W constant (Curve a), the time constant shows almost a linear relationship in the flows studied. In the case of negative step changes in heat flow rate, $-\Delta q$, W constant, "a" follows a parabolic type relationship (Curve b). What is apparent in both Curve a and Curve b is that as weight flow rate, W, increases for both $+\Delta q$ and $-\Delta q$ the time constant, "a", decreases in numerical value. Thus, at higher weight flow rates the initial slope is less in response to the same Δq. Furthermore, FIG. 36 shows that for a $+\Delta q$, the initial slope is greater than for a $-\Delta q$ for the same weight flow rate, W. On the other hand, the summary in TABLE I shows that the final ΔL_{∞} values for both $+\Delta q$ and $-\Delta q$ are in close agreement. The values listed for all runs

compare favorably with a simple steady state heat balance on this type of system. It will be noted that the values indicate that the magnitude of ΔL_{∞} in response to Δq (both positive and negative) decreases with an increase in weight flow rate, W. That section of TABLE I that is concerned with ΔW, q constant, shows basically the same type of relationship. When dealing with step changes in weight flow rate, ΔW, q constant, to determine the similarity in ΔL_{∞} the heat flow rate, q, at which the measurements were taken must be noted. This had to be varied to maintain the desired conditions (bubble-slug-annular) at the commencement and termination of each run.

As the step change in weight flow rate, ΔW, is decreased for a constant heat flow rate, q, the intial slope of the $\Delta L = \Delta L_{\infty} f(t)$ curve is changed (FIG. 37). The exception to this general form of response was found in Type 3 runs (system maintained at constant heat flow rate, q, but weight flow rate, W, increased). The temperature variation, T_{wo}, on Thermocouple 8 for a run of this nature is shown in FIG. 31. An increase in temperature can be noted for a period, followed by a reversal, and then a decrease in temperature. This would correspond to a saturated interface movement in a direction to increase the steam generating section of the vertical evaporator followed by a reversal in the direction of the movement. This reversal would cause an ultimate decrease in the length of the evaporator section. Profos [1] predicts this form of response for all run types that were studied in this investigation. He refers to movement of the commencement of evaporation counter to the

direction of flow as an "apparently paradoxical displacement." The significant number of runs that were conducted during the course of this investigation of Type 1, 2, and 4 show no indication of any displacement counter to the flow direction. It is felt that there is no such motion. The apparent reversal that was detected in the Type 3 run can be safely attributed to an inadequacy in the experimental apparatus. A needle valve was used to control the weight flow rate, W. A step change to valve position was considered to be a step change in weight flow rate, W. When such a valve is first grasped in attempting to make a change in valve position to increase flow, a depression of the valve stem takes place. This depression and the related seat movement causes a slight decrease in flow. As the valve is turned the flow rate then increases to the desired level. The increase in temperature on Thermocouple 8 is attributed to the flow disturbance resulting from valve positioning. The knee portion of this curve does resemble one that has an exponential basis. It is felt that if such initial flow disturbances could be removed, this type of run could be matched to the general form presented in equation 5.

Referring to FIGURES 10, 13, 16, 19, 22, 25, 28, 31, and 33, the relation of the various flow regimes to the total ΔT rise in response to changes in weight-flow rate, ΔW, and heat-flow rate, Δq, can be seen. The description of the flow mechanisms reported here agree with the observations of Dengler [2]. The contention that interface movement follows the same relationship as T_{wo} at Thermocouple 8 is now obvious. It can be

noted that the flow mechanisms occupy the same proportional part of each curve. The slug-annular region always occupies the knee portion of the $(1 - e^{-at})$ curve. Furthermore, as "a" decreases (signifying an increase in initial slope and a higher weight flow rate) the time of nucleation increases. The relative portion of the curves occupied by a given regime for the various weight flow rates, W, and heat flow rates, q, involved remain the same, however.

Profos [1] in his combined graphical-mathematical treatment of the dynamic behavior of forced flow evaporator systems based upon the equations of steady state conditions, derives expressions for displacement of the commencement of evaporation that are of an expotential nature. The time constant in these expressions, termed by Profos transport time and identified by the symbol, T_D, are not in numerical agreement with the time constant "a" derived in this experiment.

Profos, to simplify the basic equations which were used to describe the movement of the steam-and-water mixture in the forced-flow tube, made several assumptions. In developing his equations, the pressure in the whole system was assumed constant, the thermal loading Q of the heating surface (heat flux) was assumed to be the same over the whole evaporator system, and the mixture of steam and water was considered homogeneous at all points. Based on these assumptions, the expression

$$T_D = \frac{A\ r}{UQ\ (v'' - v')} \qquad (6)$$

was obtained. The time constants derived experimentally are not subject to the restraints of Profos. It is apparent from

the lack of agreement that any solely analytical solution of the momentum, energy, and continuity equations of forced-flow fall short in describing the dynamics of the two phase flow systems because of the assumptions that must be accepted to arrive at a workable mathematical solution.

The superheat interface movement was investigated during this series of experimental runs. It was found that T_{wo} at Thermocouple 8 followed the same general relationship developed for the saturated interface. The time constant "a" for the superheat interface movement was smaller in numerical value than that of its corresponding saturated interface movement. Therefore, the slope of superheat interface movement is less than that of saturated interface movement. Since no T_{wo} peaks are present in the superheat transition region, no experimental determination of ΔL_{∞} of the superheat interface was possible. It is felt that the movement is less than that of the saturated interface and that the variation in length of the superheater section between two steady state conditions is less than that of the evaporator section between the same two conditions. A study of the superheat interface motion is strongly dependent upon visual observation of the boiling phenomena. The use of a pyrex glass electrical conductivity tube would have assisted in the visual observation of the movement of this interface.

CHAPTER V

CONCLUSIONS

1. Saturated and superheat interface movement under transient conditions in response to step change in both weight flow rate, W, and heat flow rate, q, can be described by an equation of the general form:

$$\Delta L = \Delta L_{\infty}(1 - e^{-at})$$

2. The time constant, "a", of this equation is an indication of the time characteristics of the evaporator. It determines the initial slope of the transient response curve and itself is a strong function of weight flow rate, W, and heat flow rate, q. The slope of superheat interface transient response curve is less than that of the saturated interface response.

3. The total interface movement, ΔL_{∞}, is a function of the evaporator characteristics, weight flow rate, W, and heat flow rate, q.

4. Five regimes of boiling can be visually recognized in vertical tube evaporations: bubble, slug, slug-annular, turbulent-annular, and smooth-annular.

CHAPTER VI

RECOMMENDATIONS

1. To reduce any interference that system geometry might impose on such an experiment, a radiant heated generating element is suggested in further studies of this nature. Another possibility is to employ pyrex glass electrical conductivity tubes. With the use of such a system, the boiling phenomena under transient conditions can be visually observed. Motion pictures with a time base could be employed to trace interface movement. An analization of such a film record could yield the time and space variation desired.

2. To reduce flow disturbance at the time of step changes in weight flow rate, ΔW, a device easier to control than a hand operated needle valve should be employed.

CHAPTER VII

APPENDICES

A. DEVELOPMENT OF ANALOG TRANSFER FUNCTIONS

The problem of the development of an adequate function to describe the complexities of two-phase flow is simplified greatly if a mathematical-empirical approach is employed. In such a development, the assumptions that are resorted to will not cause serious inaccuracies in predicted results.

As discussed previously, the vertical tube evaporator can be divided into three sections: preheater, evaporator, and superheater. The outputs (temperature, pressure, mass flow rate, etc.) of each section are the inputs into the following section. A brief sketch of the manner in which the continuity equation can be solved in the preheater section follows.

Applying a control volume to the preheater section, the one-dimensional time-dependent continuity equation can be expressed

$$w_{out_e} - w_{in_e} = \frac{d}{dt}(M_e) \qquad \text{(a)}$$

in which

$$M_e = L_e \, A \, \bar{\rho}_e \qquad \text{(b)}$$

Substituting (b) into (a) and differentiating

$$w_{out_e} - w_{in_e} = A(\bar{\rho}_e \frac{dL_e}{dt} + L_e \frac{d\bar{\rho}_e}{dt}) \qquad \text{(c)}$$

The average value of a function F(x) over the interval a - b is defined by the integral

$$\bar{F}(x) = \frac{1}{b - a} \int_a^b F(x)\, dx$$

To obtain an expression for average density over the economizer

or preheater section, $\rho(L_e)$ must be integrated,

$$\bar{\rho}_e = \frac{1}{L_e} \int_0^{L_e} \rho(L_e)\, dL_e \tag{d}$$

The difference $\rho_s - \rho$ in a homogeneous fluid (which applies to the subcooled water of the preheater) is due to a corresponding temperature difference. For any liquid

$$\frac{1}{\rho} = \frac{1}{\rho_s}[1 + \beta(T_b - T_s)] \tag{e}$$

in which β is the coefficient of thermal volume expansion.

Choosing an average value of β, $\bar{\beta}$, between the design inlet and outlet conditions of the preheater, (e) can be expressed

$$\rho = \rho(L_e) = \frac{\rho_s}{[1 + \bar{\beta}(T_b - T_s)]} \tag{f}$$

in which T_b is the bulk temperature at $x = L_e$

$$T_b = T_b(L_e(t)) \tag{g}$$

The differential equation of energy conservation governing the temperature distribution in a solid body in an isotropic stationary system can be expressed as

$$[\frac{1}{r} \frac{\partial}{\partial r} (k\ r \frac{\partial T}{\partial r}) + \frac{1}{r^2}\ k\ \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2}) + W_i = \rho C \frac{\partial T}{\partial t} \tag{h}$$

The above equation assumes the internal energy change may be related to the temperature change by a specific heat.

$$de = C\, dT$$

With additional assumptions of symmetry in the θ-direction, no heat generation within the cylinder, and no temperature gradients in the z-direction, (h) reduces to

$$\left[\frac{1}{r}\frac{\partial}{\partial r}\left(k\ r\ \frac{\partial T}{\partial r}\right)\right] = \rho C\ \frac{\partial T}{\partial t} \qquad (i)$$

The solution of (i) involves the use of Bessel functions. Such a solution does not lend itself to simple analog treatment. On the other hand, assuming a negligible temperature gradient within the generating tube, which is a justifiable assumption since it is a hollow thin cylinder, a single value of temperature may be used to describe the thermal state of the tube. The energy balance for a body of arbitrary shape with negligible temperature gradients may be written

$$dQ = A_o\ h_s\ (T_g - T)dt = V\ \rho\ C\ dT \qquad (j)$$

For the generating tube under consideration, (j) can be expressed

$$2\pi\ r_o\ L\ h_s(T_g - T_{wi})dt = 2\pi(r_o - r_i)L\ \rho_{gt}\ Cp\ dT \qquad (k)$$

T_g and h_s are assumed constant. This is a reasonable assumption for the firebox of a monotube boiler. Although specific heat of a metal is a function of temperature

$$Cp = a_1 + a_2 t + a_3 t^2 + \ldots$$

Cp is assumed constant. The value for Fe varies very little from 1400°F to 2500°F (expected operating range of subcritical monotube boiler).

T	1400	1600	1800	2000	2250	2500
Cp	0.167	0.170	0.169	0.168	0.167	0.167

Expressing (k) in the form

$$r_o h_s(T_g - T_{wi})dt = (r_o - r_i)\rho_{gt}\ Cp\ d(T_g - T_{wi})$$

and integrating

$$\frac{T_g - T_{wi2}}{T_g - T_{wi1}} = \exp\left(\frac{-r_o h_s}{(r_o - r_i)\rho_{gt}\, Cp}\, t\right) \qquad (l)$$

or

$$T_{wi2} = T_g - (T_g - T_{wi1}) \exp\left(\frac{-r_o h_s}{(r_o - r_i)\rho_{gt} Cp} t\right) \qquad (m)$$

Expressing

$$b = \frac{r_o h_s}{(r_o - r_i)\rho_{gt}\, Cp}$$

and substituting into (m)

$$T_{wi2} = T_g(1 - e^{-bt}) + T_{wi1} \qquad (n)$$

The total thermal resistance, R, of the inner surface of the tube to heat transfer can be expressed

$$R = \frac{1}{h_i A_i}$$

According to the thermal circuit concept

$$Q = \frac{T_{wi} - T_b}{R}$$

or

$$Q = h_i A_i\, (T_{wi} - T_b) \qquad (o)$$

Substituting

$$A_i = 2\pi\, r_i L_e$$

and transposing, (o) can be expressed

$$T_b = T_{wi} - \frac{Q}{h_i 2\pi\, r_i L_e} \qquad (p)$$

T_{wi} and L_e are time dependent variables. T_{wi} is expressed by (n). The experimental work of this thesis showed that saturated

interface movement, ΔL, could be expressed as

$$\Delta L = \Delta L_{\infty}(1 - e^{-at})$$

or

$$L_{e_2} = L_{e_1} + \Delta L_{\infty}(1 - e^{-at}) \qquad (q)$$

The heat transfer coefficient, h_i, can be expressed by the standard McAdams expression for forced connection

$$\left(\frac{h_i}{Cp_b G}\right)\left(\frac{Cp\mu}{K}\right)_b^{0.6} = \frac{0.023}{\left(\frac{G\,D}{\mu_b}\right)^{0.2}}$$

in which subscripts refer to the various temperatures at which fluid properties are evaluated.

Substituting (q) and (n) into (p)

$$T_b = T_{wi_1} + T_g(1 - e^{-bt}) - \frac{Q}{2\pi\, r_i h_i[L_{e_1} + \Delta L_{\infty}(1 - e^{-at})]} \qquad (r)$$

the expression that was sought in (g) is obtained

$$T_b = T_b(L_e(t)) \qquad (g)$$

Substituting (r) into (f) an expression for $\rho(L_e(t))$ is obtained

$$\rho(L_e(t)) = \frac{\rho_s}{[1+\bar{\beta}\left\{T_{wi_1} + T_g(1-e^{-bt}) - \frac{Q}{2\pi\, r_i h_i[L_{e_1} + \Delta L_{\infty}(1-e^{-at})} - T_s\right\}]} \qquad (s)$$

If (s) is substituted into (d) and the results integrated an expression for $\bar{\rho}_e$ is obtained. Referring to (c)

$$w_{out_e} - w_{in_e} = A\left(\bar{\rho}_e \frac{dL_e}{dt} + L_e \frac{d\bar{\rho}_e}{dt}\right) \qquad (c)$$

it is apparent that all variables and constants required to predict w_{out_e} are available. Furthermore, they are of a nature that yields to ready placement into analog form.

B. SUMMARY OF DATA AND CALCULATIONS

TABLE II

SUMMARY OF FITTED CURVE VALUES, ΔT

Run: 1b-1

Flow Rate: 15 #/HR
Power Level Step: 591 to 1240 Watts

$$\Delta T = \Delta T_\infty (1 - e^{-at})$$

$$\Delta T_\infty = 74.0$$

$$\Delta T = 62.7 \quad t = 3.25$$

$$a = \frac{-\ln\left(\dfrac{\Delta T_\infty - \Delta T}{\Delta T_\infty}\right)}{t}$$

$$a = \frac{-\ln\left(\dfrac{74.0 - 62.7}{74.0}\right)}{3.25}$$

$$a = 0.576$$

$$\Delta T = 74.0 (1 - e^{-0.576t})$$

t	$0.576t$	$e^{-0.576t}$	$(1 - e^{-0.576t})$	ΔT
0.5	0.29	0.750	0.250	18.5
1.0	0.58	0.562	0.438	32.4
2.0	1.15	0.316	0.684	50.5
3.0	1.73	0.177	0.823	60.9
4.0	2.30	0.100	0.900	65.5
5.0	2.88	0.056	0.944	69.7
6.0	3.45	0.032	0.968	71.5
7.0	4.03	0.018	0.982	72.6
8.0	4.60	0.010	0.990	73.1

TABLE II (CONT.)

Run: 1b-1

Flow Rate: 15 #/HR
Power Level Step: 1240 to 591 Watts

$$\Delta T = \Delta T_{\infty}(1 - e^{-at})$$

$$\Delta T_{\infty} = 62.0$$

$$\Delta T = 55.4 \quad t = 4$$

$$a = \frac{-\ln\left(\frac{\Delta T_{\infty} - \Delta T}{\Delta T_{\infty}}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{62.0 - 55.4}{62.0}\right)}{4}$$

$$a = 0.56$$

$$\Delta T = 62.0\,(1 - e^{-0.56t})$$

t	0.56t	$e^{-0.56t}$	$(1 - e^{-0.56t})$	ΔT
0.5	0.28	0.756	0.244	15.2
1.0	0.56	0.571	0.429	26.6
2.0	1.12	0.326	0.674	42.0
3.0	1.68	0.189	0.814	50.5
4.0	2.24	0.106	0.894	55.4
5.0	2.80	0.062	0.938	59.1
6.0	3.36	0.035	0.965	59.8
7.0	3.92	0.019	0.981	60.9
8.0	4.47	0.011	0.989	61.2

TABLE II (CONT.)

Run: 2b

Flow Rate: 20 #/HR
Power Level Step: 584 to 1240 Watts

$$\Delta T = \Delta T_\infty (1 - e^{-at})$$

$$\Delta T_\infty = 89.7$$

$$\Delta T = 78.3 \qquad t = 4$$

$$a = \frac{-\ln\left(\frac{\Delta T_\infty - \Delta T}{\Delta T_\infty}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{89.7 - 78.3}{89.7}\right)}{4}$$

$$a = 0.516$$

$$\Delta T = 89.7\ (1 - e^{-0.516t})$$

t	$0.516t$	$e^{-0.516t}$	$1 - e^{-0.516t}$	ΔT
0.5	0.268	0.765	0.235	21.1
1.0	0.516	0.596	0.404	36.2
2.0	1.030	0.357	0.643	57.6
3.0	1.550	0.212	0.788	70.6
4.0	2.065	0.127	0.873	78.4
6.0	3.100	0.045	0.955	85.6
8.0	4.130	0.016	0.984	88.1

TABLE II (CONT.)

Run: 2b-3

Flow Rate: 20 #/HR
Power Level Step: 1240 to 584 Watts

$$\Delta T = \Delta T_\infty(1 - e^{-at})$$

$$\Delta T_\infty = 81.3$$

$$\Delta T = 76.6 \qquad t = 7$$

$$a = \frac{-\ln\left(\frac{\Delta T_\infty - \Delta T}{\Delta T_\infty}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{81.3 - 76.7}{81.3}\right)}{7}$$

$$a = 0.411$$

$$\Delta T = 81.3\ (1 - e^{-0.411t})$$

t	$0.411t$	$e^{-0.411t}$	$1 - e^{-0.411t}$	ΔT
0.5	0.205	0.814	0.185	15.0
1.0	0.411	0.662	0.338	27.4
2.0	0.822	0.439	0.561	45.6
3.0	1.230	0.292	0.708	57.5
4.0	1.645	0.193	0.807	65.5
5.0	2.055	0.128	0.872	70.9
6.0	2.460	0.085	0.915	74.4
7.0	2.880	0.056	0.944	76.6
8.0	3.285	0.037	0.963	78.4
9.0	3.700	0.025	0.975	79.1

TABLE II (CONT.)

Run: 3-1

Flow Rate: 25 #/HR
Power Level Step: 585 to 1218 Watts

$$\Delta T = \Delta T_\infty(1 - e^{-at})$$

$$\Delta T_\infty = 98.0$$

$$\Delta T = 87.7 \qquad t = 5$$

$$a = \frac{-\ln\left(\frac{\Delta T_\infty - \Delta T}{\Delta T_\infty}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{98.0 - 87.7}{98.0}\right)}{5}$$

$$a = 0.451$$

$$\Delta T = 98.0\,(1 - e^{-0.451t})$$

t	$0.451t$	$e^{-0.451t}$	$1 - e^{-0.451t}$	ΔT
0.5	0.226	0.798	0.202	19.8
1.0	0.451	0.636	0.364	35.7
2.0	0.903	0.404	0.596	58.5
3.0	1.355	0.258	0.742	72.6
4.0	1.805	0.164	0.836	82.0
5.0	2.260	0.104	0.896	88.0
6.0	2.710	0.067	0.933	91.5
7.0	3.160	0.042	0.958	94.0
8.0	3.610	0.027	0.973	95.5
9.0	4.060	0.017	0.983	96.4

TABLE II (CONT.)

Run: 3b-2

Flow Rate: 25 #/HR
Power Level Step: 1240 to 591 Watts

$$\Delta T = \Delta T_{\infty}(1 - e^{-at})$$

$$\Delta T_{\infty} = 95.6$$

$$\Delta T = 75.6 \qquad t = 4.25$$

$$a = \frac{-\ln\left(\frac{\Delta T_{\infty} - \Delta T}{\Delta T_{\infty}}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{95.6 - 75.6}{95.6}\right)}{4.25}$$

$$a = 0.369$$

$$\Delta T = 95.6\ (1 - e^{-0.369t})$$

t	$0.369t$	$e^{-0.369t}$	$1 - e^{-0.369t}$	ΔT
0.5	0.184	0.832	0.168	16.1
1.0	0.369	0.692	0.308	29.4
2.0	0.738	0.478	0.522	49.9
3.0	1.105	0.331	0.669	63.9
4.0	1.474	0.229	0.771	73.8
5.0	1.841	0.158	0.842	80.5
6.0	2.210	0.110	0.890	85.0
7.0	2.580	0.076	0.924	88.2
8.0	2.950	0.055	0.945	90.4

TABLE II (CONT.)

Run: 101b

Power Level: 986 Watts
Flow Rate Step: 25 #/HR to 16.5 #/HR

$$\Delta T = \Delta T_\infty(1 - e^{-at})$$

$$\Delta T_\infty = 12.7$$

$$\Delta T = 10.7 \qquad t = 4$$

$$a = \frac{-\ln\left(\frac{\Delta T_\infty - \Delta T}{\Delta T_\infty}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{12.7 - 10.7}{12.7}\right)}{4}$$

$$a = 0.435$$

$$\Delta T = 12.7\ (1 - e^{-0.435t})$$

t	$0.435t$	$e^{-0.435t}$	$1 - e^{-0.435t}$	ΔT
0.5	0.218	0.804	0.196	2.49
1.0	0.435	0.647	0.353	4.49
2.0	0.871	0.418	0.582	7.39
3.0	1.305	0.271	0.730	9.29
4.0	1.740	0.176	0.824	10.45
5.0	2.180	0.113	0.887	11.30
6.0	2.610	0.076	0.920	11.70
7.0	3.045	0.048	0.952	12.10
8.0	3.480	0.031	0.969	12.30
9.0	3.920	0.020	0.980	12.45

TABLE II (CONT.)

Run: 102b

Power Level: 946 Watts
Flow Rate Step: 25 #/HR to 19.5 #/HR

$$\Delta T = \Delta T_\infty(1 - e^{-at})$$

$$\Delta T_\infty = 9.7$$

$$\Delta T = 7.7 \qquad t = 3$$

$$a = \frac{-\ln\left(\frac{\Delta T_\infty - \Delta T}{\Delta T_\infty}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{9.7 - 7.7}{9.7}\right)}{3}$$

$$a = 0.526$$

$$\Delta T = 9.7\ (1 - e^{-0.526t})$$

t	$0.526t$	$e^{-0.526t}$	$1 - e^{-0.526t}$	ΔT
0.5	0.264	0.768	0.232	2.25
1.0	0.526	0.590	0.410	3.98
2.0	1.055	0.348	0.652	6.32
3.0	1.580	0.206	0.794	7.70
4.0	2.110	0.121	0.879	8.52
5.0	2.635	0.074	0.926	9.00
6.0	3.160	0.042	0.958	9.30
7.0	3.690	0.025	0.975	9.45
8.0	4.210	0.015	0.985	9.55
9.0	4.740	0.009	0.991	9.61

TABLE III

SUMMARY OF COMPUTED CURVE VALUES, ΔL

Run: 1b-1

Flow Rate: 15 #/HR
Power Level Step: 591 - 1240 Watts

$\Delta T = 74.0\ (1 - e^{-0.576t})$

$f(t) = (1 - e^{-0.576t})$ $\Delta L_{\infty} = -2.65$

$\Delta L = \Delta L_{\infty} f(t)$

$\Delta L = -2.65(1 - e^{-0.576t})$

t	$(1 - e^{-0.576t})$	ΔL
0.5	0.250	-0.665
1.0	0.438	-1.190
2.0	0.684	-1.810
3.0	0.823	-2.180
4.0	0.900	-2.385
5.0	0.944	-2.500
6.0	0.968	-2.560
7.0	0.982	-2.600
8.0	0.990	-2.620

Run: 1b-1

Flow Rate: 15 #/HR
Power Level Step: 1240 - 591 Watts

$\Delta T = 62.0\ (1 - e^{-0.56t})$ $\Delta L_{\infty} = +2.6$

$\Delta L = 2.6\ (1 - e^{-0.56t})$

t	$(1 - e^{-0.56t})$	ΔL
0.5	0.244	0.635
1.0	0.429	1.117
2.0	0.674	1.755
3.0	0.814	2.120
4.0	0.894	2.325
5.0	0.938	2.440
6.0	0.965	2.515
7.0	0.981	2.555
8.0	0.989	2.575

TABLE III (CONT.)

Run: 2b

Flow Rate: 20 #/HR
Power Level Step: 584 - 1240 Watts

$\Delta T = 89.7\ (1 - e^{-0.516t})$ $\Delta L_{\infty} = -2.5$

$\Delta L = -2.5\ (1 - e^{-0.516t})$

t	$(1 - e^{-0.516t})$	ΔL
0.5	0.235	-0.587
1.0	0.404	-1.011
2.0	0.643	-1.610
3.0	0.788	-1.972
4.0	0.873	-2.185
5.0	--	--
6.0	0.955	-2.390
7.0	--	--
8.0	0.984	-2.460

Run: 2b-3

Flow Rate: 20 #/HR
Power Level Step: 1240 - 584 Watts

$\Delta T = 81.3\ (1 - e^{-0.411t})$ $\Delta L_{\infty} = 2.50$

$\Delta L = 2.50\ (1 - e^{-0.411t})$

t	$(1 - e^{-0.411t})$	ΔL
0.5	0.185	0.464
1.0	0.338	0.846
2.0	0.561	1.405
3.0	0.708	1.775
4.0	0.807	2.025
5.0	0.872	2.185
6.0	0.915	2.290
7.0	0.944	2.360
8.0	0.963	2.410

TABLE III (CONT.)

Run: 3-1

Flow Rate: 25 #/HR
Power Level Step: 585 to 1218 Watts

$\Delta T = 98.0\ (1 - e^{-0.451t})$ $\quad \Delta L_{\infty} = -2.30$

$\Delta L = -2.30\ (1 - e^{-0.451t})$

t	$1 - e^{-0.451t}$	ΔL
0.5	0.202	-0.465
1.0	0.364	-0.836
2.0	0.596	-1.375
3.0	0.742	-1.710
4.0	0.836	-1.925
5.0	0.896	-2.065
6.0	0.933	-2.150
7.0	0.958	-2.210
8.0	0.973	-2.240

Run: 3b-2

Flow Rate: 25 #/HR
Power Level Step: 1240 to 591 Watts

$\Delta T = 95.6\ (1 - e^{-0.369t})$ $\quad \Delta L_{\infty} = 2.3$

$\Delta L = 2.3\ (1 - e^{-0.369t})$

t	$(1 - e^{-0.369t})$	ΔL
0.5	0.168	0.387
1.0	0.308	0.709
2.0	0.522	1.200
3.0	0.669	1.540
4.0	0.771	1.780
5.0	0.842	1.940
6.0	0.890	2.050
7.0	0.924	2.125
8.0	0.945	2.180

TABLE III (CONT.)

Run: 101b

Power Level: 986 Watts
Flow Rate Step: 25 #/HR to 16.5 #/HR

$\Delta T = 12.7\ (1.27\ (1 - e^{-0.345t})$ $\qquad \Delta L_{\infty} = -2.2$

$\Delta L = -2.2\ (1 - e^{-0.345t})$

t	$(1 - e^{-0.345t})$	ΔL
0.5	0.196	-0.431
1.0	0.353	-0.776
2.0	0.582	-1.281
3.0	0.730	-1.610
4.0	0.824	-1.815
5.0	0.887	-1.955
6.0	0.920	-2.030
7.0	0.952	-2.095
8.0	0.969	-2.135

Run: 102b

Power Level: 946 Watts
Flow Rate Step: 25 #/HR to 19.5 #/HR

$\Delta T = 9.7\ (1 - e^{-0.526t})$ $\qquad \Delta L_{\infty} = -2.15$

$\Delta L = -2.15\ (1 - e^{-0.526t})$

t	$(1 - e^{-0.526t})$	ΔL
0.5	0.232	-0.499
1.0	0.410	-0.881
2.0	0.652	-1.405
3.0	0.794	-1.710
4.0	0.879	-1.890
5.0	0.926	-1.995
6.0	0.958	-2.060
7.0	0.975	-2.100
8.0	0.985	-2.120

C. SAMPLE DATA AND WORK SHEETS WITH CALCULATIONS

TABLE IV

TEMPERATURE MEASUREMENTS DATA SHEET

Date: 21 Feb. 1963
Run : 3-1

Rotometer Setting = 19.5
Equivalent Flow = 25#/hr

Thermocouple Type = IC   Location = 8

Power Level = 585 Watts   V = 80   I = 7.3   Ref. Temp. = 75.0°F

	Time	Pot. Reading		Temperature	Comments	Time	Pot. Reading		Temp.
STEP	15.00	4.76	B	195.0		25.00	7.73	SA, SA	293.0
V= 117	15.25	4.98	B	202.3		26.00	7.74	SA, SA	293.3
I=10.4	15.50	5.20	B	209.7		27.00	7.73	SA, SA	293.0
P.L.=1218	15.75	5.54	B	221.0		28.00	7.74	SA, SA	293.3
WATTS	16.00	5.70	B	226.3					
	16.25	5.96	B	235.0					
	16.50	6.12	B,B	240.3					
	16.75	6.28	B,B	245.3					
	17.00	--	B,B	--					
	17.25	6.63	B,B	257.0					
	17.50	6.76	B,B,B	261.5					
	17.75	6.88	B,B,B	265.3					
	18.00	6.98	B,B,B	268.5					
	18.25	7.04	B,B,B	270.3					
	18.50	7.08	B,B,B	271.6					
	18.75	7.16	S	274.3					
	19.00	7.20	S	275.7					
	19.25	7.28	S,S	278.3					
	19.50	7.34	SA	280.3					
	19.75	7.36	SA	281.0					
	20.00	7.41	SA	282.7					
	20.25	--	--	--					
	20.50	7.48	SA,SA	285.0					
	20.75	7.50	SA,SA	285.7					
	21.00	7.52	SA,SA	286.3					
	21.25	7.54	SA,SA	287.0					
	21.50	7.56	SA,SA	287.5					
	21.75	7.58	SA,SA	288.0					
	22.00	7.60	SA,SA	288.7					
	22.25	7.64	SA,SA	290.0					
	22.50	7.64	SA,SA	290.0					
	22.75	7.65	SA,SA	290.3					
	23.00	--	SA,SA	--					
	23.25	7.66	SA,SA	290.7					
	23.50	7.70	SA,SA	292.0					
	23.75	7.71	SA,SA	292.3					
	24.00	7.68	SA,SA	289.3					
	24.25	--	SA,SA	--					
	24.50	--	SA,SA	--					
	24.75	7.72	SA,SA	292.3					

TABLE IV
TEMPERATURE MEASUREMENT DATA SHEET

TABLE V

DATE: 21 Feb. 1963

RUN = 3-1

I.C. LOCATION = 8

Power Level Step: 585 Watts to 1218 Watts

FLOW = 25#/HR

Temp. Diff Above Step

TIME	POSITIVE P.L. 585 Watts	TIME	POSITIVE P.L. 585 Watts	TIME	NEGATIVE P.L.	TIME	NEGATIVE P.L.
0.00	195.0	8.25	95.7	0.00		8.25	
0.25	7.3	8.50	97.0	0.25		8.50	
0.50	14.7	8.75	97.3	0.50		8.75	
0.75	26.0	9.00	94.3	0.75		9.00	
1.00	31.3	9.25	-	1.00		9.25	
1.25	40.0	9.50	-	1.25		9.50	
1.50	45.3	9.75	97.3	1.50		9.75	
1.75	50.0	10.00	98.0	1.75		10.00	
2.00	56.2	10.25		2.00		10.25	
2.25	62.0	10.50		2.25		10.50	
2.50	66.3	10.75		2.50		10.75	
2.75	70.3	11.00		2.75		11.00	
3.00	73.5	11.25		3.00		11.25	
3.25	75.3	11.50		3.25		11.50	
3.50	76.7	11.75		3.50		11.75	
3.75	79.3	12.00		3.75		12.00	
4.00	80.7			4.00			
4.25	83.3			4.25			
4.50	85.3			4.50			
4.75	86.0			4.75			
5.00	87.7			5.00			
5.25	-			5.25			
5.50	90.0			5.50			
5.75	90.7			5.75			
6.00	91.3			6.00			
6.25	92.0			6.25			
6.50	92.5			6.50			
6.75	93.0			6.75			
7.00	93.7			7.00			
7.25	95.0			7.25			
7.50	95.0			7.50			
7.75	95.3			7.75			
8.00	-			8.00			

TABLE V
TEMPERATURE WORK SHEET

TABLE VI

SAMPLE CALCULATION SHEET

Run: 3-1

Assuming an equation of the form

$$\Delta T = \Delta T_\infty (1 - e^{-at})$$

From WORK SHEET

$$\Delta T_{10} = \Delta T_\infty = 98.0$$

and

$$\Delta T_5 = 87.7$$

Now $\Delta T = \Delta T_\infty - \Delta T_\infty e^{-at}$

$$\frac{\Delta T_\infty - \Delta T}{\Delta T_\infty} = e^{-at}$$

$$\ln \frac{\Delta T_\infty - \Delta T}{\Delta T_\infty} = -at$$

$$a = \frac{-\ln\left(\frac{\Delta T_\infty - \Delta T}{\Delta T_\infty}\right)}{t}$$

$$a = \frac{-\ln\left(\frac{98 - 87.7}{98}\right)}{5}$$

$$a = \frac{-\ln(10.3/98.0)}{5}$$

$$a = \frac{-\ln 0.105}{5}$$

$$a = 0.451$$

and $\Delta T = 98.0\ (1 - e^{-0.451t})$

TABLE VI (CONT.)

TO OBTAIN POINTS OF COMPUTED CURVE

t	$0.451t$	$e^{-0.451t}$	$1 - e^{-0.451t}$	ΔT
0.5	0.226	0.798	0.202	19.8
1.0	0.451	0.636	0.364	35.7
2.0	0.903	0.404	0.596	58.5
3.0	1.355	0.258	0.742	72.6
4.0	1.805	0.164	0.836	82.0
5.0	2.260	0.104	0.896	88.0
6.0	2.710	0.067	0.933	91.5
7.0	3.160	0.042	0.958	94.0
8.0	3.610	0.027	0.973	95.5
9.0	4.060	0.017	0.983	96.4

D. ORIGINAL DATA

TABLE VII

TRANSIENT TEMPERATURE AND VISUAL RESPONSE
THERMOCOUPLE 8

Rotometer Setting: 13.0
Equivalent Flow: 15 #/HR
Thermocouple Type: I.C.
Location: 8

Date: 26 Feb. 1963
Run: 1b-1
Power Level Step: 591 Watts to 1240 Watts

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature °F	Δt From Step	ΔT From Step
0.00	5.96	B	235.0	0.00	0.0
0.25	6.21	B	243.3	0.25	8.3
0.50	6.40	B	249.3	0.50	14.3
0.75	6.68	S	258.7	0.75	23.7
1.00	6.94	S	267.3	1.00	32.3
1.25	7.19	S	275.0	1.25	40.0
1.50	7.24	S	277.0	1.50	42.0
1.75	7.42	SA	283.0	1.75	48.0
2.00	7.60	SA	288.7	2.00	53.7
2.25	7.67	SA	291.0	2.25	56.0
2.50	7.76	SA	294.0	2.50	59.0
2.75	7.78	SA	294.7	2.75	59.7
3.00	7.83	SA	296.3	3.00	61.3
3.25	7.87	SA	297.7	3.25	62.7
3.50	7.90	SA	298.7	3.50	63.7
3.75	7.94	SA,SA	300.0	3.75	65.0
4.00	7.96	SA,SA	300.7	4.00	65.7
4.25	7.98	SA,SA	301.3	4.25	66.3
4.50	8.03	SA,SA	302.8	4.50	67.8
4.75	--	--	--	4.75	--
5.00	8.05	SA,SA	303.3	5.00	68.3
5.25	8.07	SA,SA	304.0	5.25	69.0
5.50	8.08	SA,SA	304.3	5.50	69.3
5.75	8.08	SA,SA	304.3	5.75	69.3
6.00	8.10	SA,SA	305.0	6.00	70.0
6.25	8.14	SA,SA	306.3	6.25	71.3
6.50	8.13	SA,SA	306.0	6.50	71.0
6.75	8.14	SA,SA	306.3	6.75	71.3
7.00	8.13	SA,SA	306.0	7.00	71.0
7.25	8.15	SA,SA	306.7	7.25	71.7
7.50	8.16	SA,SA	307.0	7.50	72.0
7.75	8.16	SA,SA	307.0	7.75	72.0
8.00	8.15	SA,SA	306.7	8.00	71.7
8.25	8.17	SA,SA	307.3	8.25	72.3
8.50	8.18	SA,SA	307.7	8.50	72.7
8.75	--	--	--	8.75	--
9.00	8.22	SA,SA	309.0	9.00	74.0
9.50	8.22	SA,SA	309.0	9.50	74.0
10.00	8.22	SA,SA	309.0	10.00	74.0

TABLE VII (CONT.)

Rotometer Setting: 13.0
Equivalent Flow: 15 #/HR
Thermocouple Type: I.C.
Location: 8

Date: 26 Feb. 1963
Run: 1b-1
Power Level Step: 1240 Watts to 591 Watts

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature °F	Δt From Step	ΔT From Step
16.50	8.27	SA,SA,SA	310.7	0.00	0.0
16.75	8.09	SA,SA,SA	304.7	0.25	6.0
17.00	7.80	SA,SA,SA	295.3	0.50	15.4
17.25	7.57	SA,SA	287.7	0.75	23.0
17.50	7.32	SA,SA	279.7	1.00	31.0
17.75	7.29	SA,SA	278.7	1.25	32.0
18.00	7.17	SA,SA	274.7	1.50	36.0
18.25	7.08	SA,SA	271.7	1.75	39.0
18.50	6.94	SA,SA	267.3	2.00	43.4
18.75	6.88	SA	265.3	2.25	45.4
19.00	6.82	SA	263.4	2.50	47.3
19.25	6.71	SA	259.7	2.75	51.0
19.50	6.71	SA	259.7	3.00	51.0
19.75	6.66	SA	258.0	3.25	52.7
20.00	6.62	SA	256.7	3.50	54.0
20.25	6.58	SA	255.3	3.75	55.4
20.50	6.58	SA	255.3	4.00	55.4
20.75	6.56	SA	254.7	4.25	56.0
21.00	6.55	S,S	254.3	4.50	56.4
21.25	6.54	S,S	254.0	4.75	56.7
21.50	6.52	S,S	253.3	5.00	57.4
21.75	6.50	S,S	252.7	5.25	58.0
22.00	6.47	S	251.7	5.50	59.0
22.25	6.49	S	252.3	5.75	58.4
22.50	--	S	--	6.00	--
22.75	6.44	S	250.7	6.25	60.0
23.00	6.46	S	251.3	6.50	59.4
23.25	6.46	S	251.3	6.75	59.4
23.50	--	S	--	7.00	--
23.75	--	S	--	7.25	--
24.00	6.42	S	250.0	7.50	60.7
24.25	--	S	--	7.75	--
24.50	6.42	B	250.0	8.00	60.7
24.75	--	B	--	8.25	--
25.00	6.42	B	250.0	8.50	60.7
25.25	6.38	B	248.7	8.75	62.0
25.50	6.37	B	248.7	9.00	62.0

TABLE VII (CONT.)

Rotometer Setting: 16.5
Equivalent Flow: 20 #/HR
Thermocouple Type: I.C.
Location: 8

Date: 26 Feb. 1963
Run: 2b
Power Level Step: 584 Watts to 1240 Watts

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature °F	Δt From Step	ΔT From Step
39.00	5.36	B	215.0	0.00	0.0
39.25	5.62	B	223.7	0.25	8.7
39.50	5.98	BB	235.7	0.50	20.7
39.75	6.18	BB	242.3	0.75	27.3
40.00	6.40	BB	249.3	1.00	34.0
40.25	6.62	BBB	256.7	1.25	41.7
40.50	6.82	BBB	263.3	1.50	48.3
40.75	7.13	BBB	273.3	1.75	53.5
41.00	7.22	S	276.3	2.00	58.3
41.25	7.32	S	279.7	2.25	64.7
41.50	7.40	SA	282.3	2.50	67.3
41.75	7.52	SA	286.3	2.75	70.3
42.00	7.49	SA	285.3	3.00	72.5
42.25	7.64	SA	290.0	3.25	75.0
42.50	7.67	SA	291.0	3.50	76.0
42.75	--	SA	--	3.75	--
43.00	7.74	SA,SA	293.3	4.00	78.3
43.25	--	SA,SA	--	4.25	--
43.50	7.80	SA,SA	295.3	4.50	80.3
43.75	--	SA,SA	--	4.75	--
44.00	7.85	SA,SA	297.0	5.00	82.0
44.25	--	SA,SA	--	5.25	--
44.50	7.88	SA,SA	298.0	5.50	83.0
44.75	--	SA,SA	--	5.75	--
45.00	7.92	SA,SA	299.3	6.00	84.3
45.25	7.94	SA,SA	300.0	6.25	85.0
45.50	7.96	SA,SA	300.7	6.50	85.7
45.75	7.94	SA,SA	300.0	6.75	85.0
46.00	--	SA,SA	--	7.00	--
46.25	7.98	SA,SA	301.0	7.25	86.3
46.50	--	SA,SA	--	7.50	--
46.75	--	SA,SA	--	7.75	--
47.00	8.00	SA,SA	302.0	8.00	87.0
48.00	8.02	SA,SA	302.5	9.00	87.5
49.00	8.05	SA,SA	303.3	10.00	88.3
50.00	8.08	SA,SA	304.3	11.00	89.3
51.00	8.09	SA,SA	304.7	12.00	89.7

TABLE VII (CONT.)

Rotometer Setting: 16.5
Equivalent Flow: 20 #/HR
Thermocouple Type: I.C.
Location: 8

Date: 26 Feb. 1963
Run: 2b-3
Power Level Step: 1240 Watts to 584 Watts

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature °F	Δt From Step	ΔT From Step
66.00	8.20	SA	308.3	0.00	0.0
66.25	8.02	SA	302.0	0.25	5.8
66.50	7.70	SA	292.0	0.50	16.3
66.75	7.54	SA	287.0	0.75	21.3
67.00	7.32	SA	279.7	1.00	28.6
67.25	7.20	SA	275.7	1.25	32.6
67.50	7.04	S	270.3	1.50	38.0
67.75	6.98	S	268.5	1.75	39.8
68.00	6.85	S	264.3	2.00	44.0
68.25	6.76	S	261.3	2.25	47.0
68.50	6.68	S	258.7	2.50	49.6
68.75	6.61	BBB	256.3	2.75	52.0
69.00	6.54	BBB	254.0	3.00	54.3
69.25	--	BBB	--	3.25	--
69.50	6.41	BB	249.6	3.50	58.7
69.75	6.35	BB	247.6	3.75	60.7
70.00	6.30	B	246.0	4.00	62.3
70.25	6.23	B	243.8	4.25	64.5
70.50	6.18	B	242.3	4.50	66.0
70.75	--	B	--	4.75	--
71.00	6.07	B	238.6	5.00	69.7
71.25	--	B	--	5.25	--
71.50	6.04	B	237.7	5.50	70.6
71.75	--	B	--	5.75	--
72.00	5.94	B	243.3	6.00	65.0
72.25	--	B	--	6.25	--
72.50	5.90	B	233.0	6.50	75.3
72.75	--	B	--	6.75	--
73.00	5.86	B	231.7	7.00	76.6
73.25	--	B	--	7.25	--
73.50	5.82	B	230.3	7.50	78.0
73.75	--	B	--	7.75	--
74.00	5.82	B	230.3	8.00	78.0
74.25	5.72	B	227.0	9.25	81.3

TABLE VII (CONT.)

Rotometer Setting: 19.5
Equivalent Flow: 25 #/HR
Thermocouple Type: I.C.
Location: 8

Date: 21 Feb. 1963
Run: 3-1
Power Level Step: 585 Watts to 1218 Watts

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature ^{o}F	Δt From Step	ΔT From Step
15.00	4.76	B	195.0	0.00	0
15.25	4.98	B	202.3	0.25	7.3
15.50	5.20	B	209.7	0.50	14.7
15.75	5.54	B	221.0	0.75	26.0
16.00	5.70	B	226.3	1.00	31.3
16.25	5.96	B	235.0	1.25	40.0
16.50	6.12	B,B	240.3	1.50	45.3
16.75	6.28	B,B	245.3	1.75	50.0
17.00	--	B,B	--	2.00	56.2
17.25	6.63	B,B	257.0	2.25	62.0
17.50	6.76	B,B,B	261.3	2.50	66.3
17.75	6.88	B,B,B	265.3	2.75	70.3
18.00	6.98	B,B,B	268.5	3.00	73.5
18.25	7.04	B,B,B	270.3	3.25	75.3
18.50	7.08	B,B,B	271.6	3.50	76.7
18.75	7.16	S	274.3	3.75	79.3
19.00	7.20	S	275.7	4.00	80.7
19.25	7.28	S,S	278.3	4.25	83.3
19.50	7.34	SA	280.3	4.50	85.3
19.75	7.36	SA	281.0	4.75	86.0
20.00	7.41	SA	282.7	5.00	87.7
20.25	--	--	--	5.25	--
20.50	7.48	SA,SA	285.0	5.50	90.0
20.75	7.50	SA,SA	285.7	5.75	90.7
21.00	7.52	SA,SA	286.3	6.00	91.3
21.25	7.54	SA,SA	287.0	6.25	92.0
21.50	7.56	SA,SA	287.5	6.50	92.5
21.75	7.58	SA,SA	288.0	6.75	93.0
22.00	7.60	SA,SA	288.7	7.00	93.7
22.25	7.64	SA,SA	290.0	7.25	95.0
22.50	7.64	SA,SA	290.0	7.50	95.0
22.75	7.65	SA,SA	290.3	7.75	95.3
23.00	--	--	--	8.00	--
23.25	7.66	SA,SA	290.7	8.25	95.7
23.50	7.70	SA,SA	292.0	8.50	97.0
23.75	7.71	SA,SA	292.3	8.75	97.3
24.00	7.68	SA,SA	289.3	9.00	94.3

TABLE VII (CONT.)

Rotometer Setting: 19.5
Equivalent Flow: 25 #/HR
Thermocouple Type: I.C.
Location: 8

Date: 25 Feb. 1963
Run: 3b-2
Power Level Step: 1240 Watts to 591 Watts

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature °F	Δt From Step	ΔT From Step
32.00	8.14	SA	306.3	0.00	0
32.25	7.89	SA	298.3	0.25	8.0
32.50	7.65	S	290.3	0.50	16.0
32.75	7.50	S	285.7	0.75	20.6
33.00	7.18	S	275.0	1.00	31.3
33.25	7.10	S	272.3	1.25	34.0
33.50	6.94	BBB	267.3	1.50	39.0
33.75	--	BB	--	1.75	--
34.00	6.67	B	258.3	2.00	48.0
34.25	6.56	B	254.7	2.25	51.6
34.50	6.48	B	252.0	2.50	54.3
34.75	6.32	B	246.7	2.75	59.6
35.00	6.18	B	242.3	3.00	64.0
35.25	--	-	--	3.25	--
35.50	6.04	B	237.7	3.50	68.6
35.75	5.94	B	234.3	3.75	72.0
36.00	--	-	--	4.00	--
36.25	5.83	B	230.7	4.25	75.6
36.50	5.74	B	227.7	4.50	78.6
36.75	--	B	--	4.75	--
37.00	5.70	B	226.3	5.00	80.0
37.25	--	B	--	5.25	--
37.50	5.59	B	222.7	5.50	83.6
37.75	--	B	--	5.75	--
38.00	5.53	B	220.7	6.00	85.6
38.25	--	B	--	6.25	--
38.50	5.50	B	219.7	6.50	86.6
38.75	--	B	--	6.75	--
39.00	5.44	B	217.7	7.00	88.7
39.25	--	B	--	7.25	--
39.50	5.44	B	217.7	7.50	88.6
39.75	--	B	--	7.75	--
40.00	5.40	B	216.3	8.00	90.0
41.00	5.34	B	214.3	9.00	92.0
42.50	5.27	B	212.0	10.50	94.3
43.00	5.25	B	211.3	11.00	95.0
44.00	5.23	B	210.7	12.00	95.6
45.00	5.23	B	210.7	13.00	95.6

TABLE VII (CONT.)

Power Level: 986 Watts
Thermocouple Type: I.C.
Location: 8

Date: 2 March 1963
Run: 101b
Flow Step: From 25 #/HR to 16.5 #/HR

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature ^{o}F	Δt From Step	ΔT From Step
0.00	7.89	B	298.3	0.00	0
0.25	7.92	B	299.3	0.25	1.0
0.50	7.96	B	300.5	0.50	2.2
0.75	7.99	B	301.5	0.75	3.2
1.00	8.02	S	302.5	1.00	4.2
1.25	8.08	S	304.3	1.25	6.0
1.50	8.09	S	304.6	1.50	6.3
1.75	8.12	S	305.7	1.75	7.4
2.00	8.14	S	306.3	2.00	8.0
2.25	8.14	SA	306.3	2.25	8.0
2.50	--	SA	--	2.50	--
2.75	8.16	SA	307.0	2.75	8.7
3.00	8.18	SA	307.7	3.00	9.0
3.25	8.20	SA	308.3	3.25	10.0
3.50	8.20	SA	308.3	3.50	10.0
3.75	8.22	SA	309.0	3.75	10.7
4.00	8.22	SA	309.0	4.00	10.7
4.25	8.22	SA	309.0	4.25	10.7
4.50	8.23	SA	309.3	4.50	11.0
4.75	--	--	--	4.75	--
5.00	8.24	SA	309.7	5.00	11.4
5.25	8.24	SA	309.7	5.25	11.4
5.50	8.24	SA	309.7	5.50	11.4
5.75	--	--	--	5.75	--
6.00	8.25	SA	310.0	6.00	11.7
6.25	8.25	SA	310.0	6.25	11.7
6.50	8.26	SA	310.3	6.50	12.0
6.75	8.26	SA	310.3	6.75	12.0
7.00	8.26	SA	310.3	7.00	12.0
7.25	--	--	--	7.25	--
7.50	8.26	SA	310.3	7.50	12.0
7.75	8.26	SA	310.3	7.75	12.0
8.00	--	--	--	8.00	--
8.25	8.27	SA	310.7	8.25	12.4
8.50	--	--	--	8.50	--
8.75	8.27	SA	310.7	8.75	12.4
9.00	8.28	SA	311.0	9.00	12.7
9.50	8.27	SA	310.7	9.50	12.4

TABLE VII (CONT.)

Power Level: 966 Watts
Thermocouple Type: I.C.
Location: 8

Date: 11 March 1963
Run: 101c
Flow Step: From 16.5 #/HR to 25 #/HR

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature ^oF	Δt From Step	ΔT From Step
0.00	8.12	SA	305.7	0.00	0.0
0.25	8.13	SA	306.0	0.25	+0.3
0.50	8.13	SA	306.0	0.50	+0.3
0.75	8.13	SA	306.0	0.75	+0.3
1.00	8.13	SA	306.0	1.00	+0.3
1.25	8.11	S	305.3	1.25	0.4
1.50	8.10	S	305.0	1.50	0.7
1.75	8.09	S	304.7	1.75	1.0
2.00	8.06	S	303.7	2.00	2.0
2.25	--	B	--	2.25	--
2.50	8.02	B	302.5	2.50	3.2
2.75	--	B	--	2.75	--
3.00	7.98	B	301.3	3.00	4.4
3.25	--	B	--	3.25	--
3.50	7.96	B	300.7	3.50	5.0
3.75	7.94	B	300.0	3.75	5.7
4.00	7.94	B	300.0	4.00	5.7
4.25	7.91	B	299.0	4.25	6.7
4.50	7.90	B	298.7	4.50	7.0
4.75	7.90	B	298.7	4.75	7.0
5.00	7.88	B	298.0	5.00	7.7
5.25	7.87	B	297.7	5.25	8.0
5.50	7.86	B	297.3	5.50	8.4
5.75	7.86	B	297.3	5.75	8.4
6.00	7.84	B	296.7	6.00	9.0
6.25	7.85	B	297.0	6.25	8.7
7.00	7.83	B	296.3	7.00	9.4
8.00	7.81	B	295.7	8.00	10.0
9.00	7.80	B	295.3	9.00	10.4
10.00	7.80	B	295.3	10.00	10.4
11.00	7.80	B	295.3	11.00	10.4
12.00	7.80	B	295.3	12.00	10.4

TABLE VII (CONT.)

Power Level: 946 Watts
Thermocouple Type: I.C.
Location: 8

Date: 2 March 1963
Run: 102b
Flow Step: 25 #/HR to 19.5 #/HR

t (Minutes)	Potentiometer (M.V.)	Visual Code	Temperature ^{o}F	Δt From Step	ΔT From Step
0.00	7.98	B	301.3	0.00	0.0
0.25	8.00	B	302.0	0.25	0.8
0.50	8.05	B	303.3	0.50	2.0
0.75	8.08	B	304.3	0.75	3.0
1.00	8.11	B	305.3	1.00	4.0
1.25	8.13	S	306.0	1.25	4.7
1.50	8.15	S	306.7	1.50	5.4
1.75	8.15	S	306.7	1.75	5.4
2.00	8.17	S	307.3	2.00	6.0
2.25	8.19	S	308.0	2.25	6.7
2.50	--	S	--	2.50	--
2.75	--	S	--	2.75	--
3.00	8.22	SA	309.0	3.00	7.7
3.25	--	SA	--	3.25	--
3.50	8.24	SA	309.7	3.50	8.4
3.75	8.23	SA	309.3	3.75	8.0
4.00	8.24	SA	309.7	4.00	8.4
4.25	8.24	SA	309.7	4.25	8.4
4.50	8.25	SA	310.0	4.50	8.7
4.75	--	--	--	4.75	--
5.00	8.26	SA	310.3	5.00	9.0
5.25	8.27	SA	310.7	5.25	9.4
5.50	9.27	SA	310.7	5.50	9.4
5.75	--	--	--	5.75	--
6.00	8.26	SA	310.3	6.00	9.0
6.25	8.27	SA	310.7	6.25	9.4
6.50	8.27	SA	310.7	6.50	9.4
6.75	8.26	SA	310.3	6.75	9.0
7.00	8.27	SA	310.7	7.00	9.4
7.25	--	--	--	7.25	--
7.50	8.27	SA	310.7	7.50	9.4
7.75	--	--	--	7.75	--
8.00	8.28	SA	311.0	8.00	9.7
8.25	--	--	--	8.25	--
8.50	8.28	SA	311.0	8.50	9.7
8.75	8.28	SA	311.0	8.75	9.7
9.00	8.28	SA	311.0	9.00	9.7
10.00	8.28	SA	311.0	10.00	9.7

TABLE VIII

TEMPERATURE VARIATION THERMOCOUPLES 1 - 7 AT STEADY STATE CONDITIONS FOR EACH RUN

Run: 1b-1

Before +Δq

Thermocouple	M.V.	T(°F)	ΔT
1	2.36	135.5	0
2	3.12	166.5	31.0
3	3.26	172.5	37.0
4	3.38	177.0	41.5
5	3.49	181.5	46.0
6	3.59	185.5	50.0
7	3.64	187.5	52.0

After +Δq

Thermocouple	M.V.	T(°F)	ΔT
1	3.88	197.0	0
2	5.63	263.0	66.0
3	5.95	275.0	78.0
4	6.09	280.0	83.0
5	6.10	280.5	83.5
6	5.84	271.0	74.0
7	5.41	255.0	58.0

Run: 2b

Before +Δq

Thermocouple	M.V.	T(°F)	ΔT
1	2.38	136.5	0
2	2.94	159.5	23.0
3	3.08	165.0	28.5
4	3.17	168.5	32.0
5	3.24	171.5	35.0
6	3.23	171.0	34.5
7	3.24	171.5	35.0

After +Δq

Thermocouple	M.V.	T(°F)	ΔT
1	3.87	196.5	0
2	5.21	247.5	51.0
3	5.51	258.5	62.0
4	5.67	264.5	68.0
5	5.69	265.5	69.0
6	5.51	256.5	60.0
7	5.05	241.5	45.0

TABLE VIII (CONT.)

Run: 3-1

Before $+\Delta q$

Thermocouple	M.V.	T($^{\circ}$F)	ΔT
1	2.39	137.0	0.0
2	2.84	155.0	18.0
3	2.96	160.0	23.0
4	3.03	163.0	26.0
5	3.04	163.5	26.5
6	3.13	167.0	30.0
7	3.13	167.0	30.0

After $+\Delta q$

Thermocouple	M.V.	T($^{\circ}$F)	ΔT
1	3.82	194.5	0.0
2	4.99	239.5	45.0
3	5.24	249.6	55.1
4	5.43	255.5	61.0
5	5.45	256.5	62.0
6	5.35	252.5	58.0
7	5.08	242.5	48.0

TABLE VIII (CONT.)

Run: 1b-1

Before $-\Delta q$

Thermocouple	M.V.	T($^{\circ}$F)	ΔT
1	3.89	197.0	0.0
2	5.44	256.0	59.0
3	5.74	267.0	70.0
4	5.86	271.5	74.5
5	5.88	272.0	75.0
6	5.74	267.0	70.0
7	5.41	255.0	58.0

After $-\Delta q$

Thermocouple	M.V.	T($^{\circ}$F)	ΔT
1	2.44	139.0	0.0
2	3.23	171.0	32.0
3	3.43	179.0	40.0
4	3.61	186.0	47.0
5	3.73	190.5	51.5
6	3.77	192.5	53.5
7	3.84	195.0	56.0

Run: 2b-3

Before $-\Delta q$

Thermocouple	M.V.	T($^{\circ}$F)	ΔT
1	3.87	196.5	0.0
2	5.29	250.5	54.0
3	5.63	263.0	66.5
4	5.80	269.5	73.0
5	5.83	270.5	74.0
6	5.58	261.0	69.5
7	5.11	243.5	47.0

After $-\Delta q$

Thermocouple	M.V.	T($^{\circ}$F)	ΔT
1	2.39	137.0	0.0
2	3.03	163.0	26.0
3	3.17	168.5	31.5
4	3.33	175.0	38.0
5	3.38	177.0	40.0
6	3.47	180.5	43.5
7	3.49	181.5	44.5

TABLE VIII (CONT.)

Run: 3b-2

Before $-\Delta q$

Thermocouple	M.V.	T(^{o}F)	ΔT
1	3.81	194.0	0.0
2	4.93	237.0	43.0
3	5.27	249.5	55.5
4	5.42	255.0	61.0
5	5.47	257.0	63.0
6	5.36	253.0	59.0
7	5.07	242.0	48.0

After $-\Delta q$

Thermocouple	M.V.	T(^{o}F)	ΔT
1	2.36	135.5	0.0
2	2.82	154.5	19.0
3	2.96	160.0	24.5
4	3.07	164.5	29.0
5	3.09	165.5	30.0
6	3.14	167.5	32.0
7	3.14	167.5	32.0

TABLE VIII (CONT.)

Run: 101b

Before-ΔW

Thermocouple	M.V.	T(^{o}F)	ΔT
1	3.12	166.5	0.0
2	3.88	196.5	30.0
3	4.06	203.5	37.0
4	4.16	207.5	41.0
5	4.21	209.5	43.0
6	4.24	210.5	44.0
7	4.24	210.5	44.0

After -ΔW

Thermocouple	M.V.	T(^{o}F)	ΔT
1	3.26	172.0	0.0
2	4.30	214.0	42.0
3	4.65	226.5	54.5
4	4.85	234.0	62.0
5	4.85	234.0	62.0
6	4.80	232.0	60.0
7	4.56	223.0	51.0

Run: 102b

Before -ΔW

Thermocouple	M.V.	T(^{o}F)	ΔT
1	3.24	171.5	0.0
2	4.24	210.5	39.0
3	4.51	221.0	49.5
4	4.70	228.5	57.0
5	4.80	231.5	60.0
6	4.83	233.0	61.5
7	4.83	233.0	61.5

After -ΔW

Thermocouple	M.V.	T(^{o}F)	ΔT
1	3.27	172.5	0.0
2	4.58	223.5	51.0
3	4.97	238.5	66.0
4	5.24	248.5	76.0
5	5.32	251.5	79.0
6	5.13	244.5	72.0
7	4.46	219.0	46.5

TABLE VIII (CONT.)

Run: 101c

Before +ΔW

Thermocouple	M.V.	T(°F)	ΔT
1	3.26	172.0	0.0
2	4.40	217.0	45.0
3	4.68	227.5	55.5
4	4.86	234.0	62.0
5	4.87	234.5	62.5
6	4.75	230.0	58.0
7	4.47	219.5	47.5

After +ΔW

Thermocouple	M.V.	T(°F)	ΔT
1	3.21	170.5	0.0
2	4.03	202.5	32.5
3	4.25	211.0	40.5
4	4.39	216.5	46.0
5	4.50	220.5	50.0
6	4.50	220.5	50.0
7	4.50	220.5	50.0

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LITERATURE CITATIONS

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G. TABLE OF SYMBOLS

TABLE OF SYMBOLS

A	=	inside cross-sectional area of tube, ft^2
A_i	=	inside heat transfer area of tube, ft^2
A_o	=	outside heat transfer area of tube, ft^2
a	=	experimental time constant, $min.^{-1}$
B	=	bubble regime of boiling, repeating symbol signifies increasing intensity of phenomenon
C	=	specific heat, BTU lb_m^{-1} $^oF^{-1}$
Cp	=	specific heat at constant pressure, BTU lb_m^{-1} $^oF^{-1}$
D	=	inside diameter of generating tube, ft.
G	=	mass flow rate per unit area, lb_m $hr^{-1}ft^{-2}$
h_i	=	forced-convection heat transfer coefficient, BTU hr^{-1} ft^{-2} $^oF^{-1}$
h_s	=	firebox tube scale heat transfer coefficient, BTU hr^{-1} ft^{-2} $^oF^{-1}$
K	=	thermal conductivity, BTU hr^{-1} ft^{-1} $^oF^{-1}$
L	=	length of generating tube, ft
Lp	=	length of generating tube at which peak outside wall temperature occurs, ft.
M	=	mass, lb_m
Q	=	heat flux for inside surface of tube, BTU hr^{-1} ft^{-2}
q	=	heat flow rate, WATTS
r	=	latent heat of evaporation, BTU lb_m^{-1}
r_i	=	inside tube radius, ft
r_o	=	outside tube radius, ft
S	=	slug regime of boiling, repeating symbol signifies increasing intensity of phenomenon
SA	=	slug-annular regime of boiling, repeating symbol signifies increasing intensity of phenomenon

T	=	temperature, ^{o}F
t	=	time, hr, min, sec
T_D	=	transport time, min^{-1}
TA	=	turbulent-annular regime of boiling, repeating symbol signifies increasing intensity of phenomenon
U	=	inside circumference of tube, ft
V	=	volume of generating tube, ft^3
v'	=	specific volume of water at boiling point, $ft^3\ lb_m^{-1}$
v"	=	specific volume of saturated steam, $ft^3\ lb_m^{-1}$
W	=	weight flow rate, $lb_f\ hr^{-1}$
w	=	mass flow rate, $lb_m\ hr^{-1}$
W_i	=	internal heat generation per unit volume and time, BTU $ft^{-3}\ hr^{-1}$

GREEK

β	=	coefficient of thermal volume expansion, $^{o}F^{-1}$
$\overline{\beta}$	=	average coefficient of thermal volume expansion, $^{o}F^{-1}$
μ	=	viscosity, $lb_m\ hr^{-1}\ ft^{-1}$
ρ	=	density, $lb_m\ ft^{-3}$
$\overline{\rho}$	=	average density, $lb_m\ ft^{-3}$

SUBSCRIPTS

b	=	bulk
e	=	preheater, economizer
g	=	gas
gt	=	generating tube
i	=	inside
o	=	outside
p	=	peak

s	=	at inlet of preheater section
w	=	wall
x	=	variable distance from inlet of preheater section
∞	=	final steady state value after step change
1	=	steady state value before step change
2	=	transient value after step change

PREFIX

Δ	=	increment, step change